NUMERO: **2518A** ANNO: 2021

A P P U N T I

STUDENTE: Sobrero Giovanni

MATERIA: Powertrain Components Design, vol. I, Engine - Prof. Delprete

Powertrain Components Design

Professors:

Engine Lectures: Prof. Cristiana Delprete
(Full Professor of Design and Construction of Machines, DIMEAS, 3rd floor, 011.090.7747, cristiana.delprete@polito.it)

Transmission Lectures: Prof. Enrico Galvagno and Prof. Carlo Baret
(Associate Professor of Applied Mechanics, DIMEAS, 4th floor, 011.090.6928, enrico.galvagno@polito.it) (Adjunct Professor carlo.baret@gmail.com)

Engine and Transmission Practices: Prof. Carlo Rosso
(Associate Professor of Design and Construction of Machines, DIMEAS, 4th floor, 011.090.5817, carlo.rosso@polito.it)

Giovanni Sobrero' Schemes

A.A. 2019 – 2020 (10 cfu)

ENGINE – Program:

E1. Crank mechanism
E2. Wrist pin
E3. Connecting rod
E4. Crankshaft
E5. Main bearings
E6. Piston
E7. Cylinder head
E8. Cylinder block
E9. Oil pan
E10. Thermo-mechanical fatigue
E11. Exhaust manifold
E12. (Timing system)

E – Notes and Textbooks:

Lectures modules are available on the Corse Website;

Reference books for study improvement:

• Makartchouk A., Diesel Engine Engineering, ISBN: 0-8247- 0702-8, Marcel Dekker Inc., New York, NY, USA, 2002;
• Manning J., Internal Combustion Engine Design, ISBN 978-0- 9573292-0-1, Ricardo UK Limited, UK, 2012;
• Hoag K.L., Vehicular Engine Design, ISBN: 0-7680-1661-4, SAE International, Warrendale, PA, USA, 2006;
• Stone R., Introduction to Internal Combustion Engines, ISBN 0-7680-0495-0, SAE International, Warrendale, PA, USA, 1999;
• Taylor C.F., The internal-Combustion Engine in Theory and Practice, The M.I.T Press, Cambridge, UK, 1997.

TRASMISSION– Program:

T0. Introduction
T1. Transmission functions and architectures
T2. Gear ratios definition
T3. Manual transmissions: configurations, efficiency
T4. Manual transmissions: cars
T5. Manual transmissions: industrial vehicles
T6. Mission, load profiles and components development process
T7. Components design and testing: gears
T8. Components design and testing: other components
T9. Synchronizers
T10. Start-up devices for manual transmissions
T11. DCU and DCT
T12. Differentials and final drives (part 1)
T13. Differentials and final drives (part 2)
T14. Shafts and joints
T15. Shifting mechanisms
T16. Start-up devices for automatic transmissions
T17. Automatic transmissions: fixed axis (part 1)
T18. Automatic transmissions: epicycloidal (part 2)
T19. Automatic transmissions: CVT, AT for I.V. (part 3)
T20. Transmissions for HEV and FEV

T – Textbooks:

Course Textbook: G. Geeta, L. Morello "The Automotive Chassis – Vol. I and II", Springer, 2009;
Other Books: G. Lechner, H. Naunheimer, "Automotive Transmissions, Fundamentals, Selection, Design and Application", Springer, Berlin, 1999.

E & T – Exam:

The exam is an **oral exam** and it is constituted by:

- **Presentation** the students made at the **practice;**
- Discussion (about **30 min**) of **2 questions on Engine** topics and **2 questions on Transmission** topics.

Each element is **separately evaluated** (score in thirtieths) and the **overall score** is defined by:

- 1/3 practice presentation, 1/3 engine topics discussion, and 1/3 transmission topics discussion.

The oral discussion on engine and transmission topics intends understanding the actual level of comprehension of all course:

- **Schemes and formulas** must be **demonstrated and discussed;**
- **Methodological steps** of topic development must be **highlighted and explained.**

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POWERTRAIN COMPONENTS DESIGN (10 CFU) (2020)

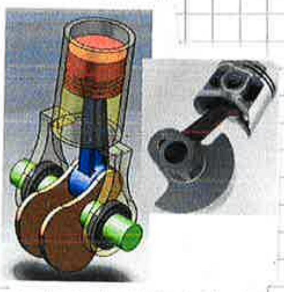
ENGINE: PROF. CRISTIANA DELPRETE
TRANSMISSION: PROF. ENRICO GALVAGNO

SCHEMES OF GIOVANNI SOBRERO

ENGINE - PROGRAM:

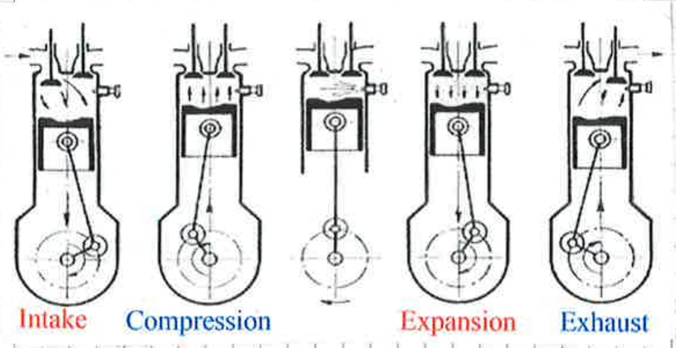
- E1 CRANK MECHANISM
- E2 WRIST PIN
- E3 CONNECTING ROD
- E4 CRANKSHAFT
- E5 MAIN BEARINGS
- E6 PISTON
- E7 CYLINDER HEAD
- E8 CYLINDER BLOCK
- E9 OIL PAN
- E10 THERMO-MECHANICAL FATIGUE (TMF)
- E11 EXHAUST MANIFOLD
- E12 TIMING SYSTEM (NOT IN 2019/2020 PROGRAM)

E1 CRANK MECHANISM



- INTRODUCTION
- LAYOUT
- SYMBOLS
- KINEMATICS
  - K - PISTON DISPLACEMENT, CENTERED
  - K - PISTON DISPLACEMENT, OFFSET
  - K - PISTON DISPLACEMENT, OFFSET VS CENTERED
  - K - PISTON VELOCITY, OFFSET
  - K - PISTON VELOCITY, CENTERED
  - K - APPROXIMATED PISTON VELOCITY, CENTERED
  - K - PISTON VELOCITY, OFFSET VS CENTERED (APPROX)
  - K - MEAN PISTON VELOCITY, CENTERED
  - K - PISTON ACCELERATION, OFFSET
  - K - PISTON ACCELERATION, CENTERED
  - K - APPROXIMATED PISTON ACCELERATION, CENTERED
  - K - PISTON ACCELERATION, OFFSET VS CENTERED (APPROX.)
  - K - APPROXIMATED PISTON ACCELERATION, CENTERED - DEEPENING
  - K - CONNECTING ROD ANGULAR VELOCITY AND ACCELERATION
- DYNAMICS
  - D - FORCES ON THE CRANK MECHANISM
  - D - CONNECTING ROD REDUCTION
  - D - RECIPROCATING INERTIAL FORCE
  - D - CENTRIFUGAL INERTIAL FORCE
  - D - RESULTANT FORCE ON THE CRANK MECHANISM
  - D - FORCES ON THE CONNECTING ROD
  - D - TORQUE ON THE SINGLE CRANK
  - D - FORCES AND MOMENT ON THE CYLINDER BLOCK
- MULTI-CYLINDER ENGINE AND MOST STRESSED CRANK
- ENGINE DEGREE OF IRREGULARITY

- INTAKE AND EXPANSION STROKES BELONG TO **DOWNHILL** (OR INWARD) ("DISCESA") MOVEMENTS, COMPRESSION AND EXHAUST STROKES TO **UPHILL** (OR OUTWARD) ("SALITA") MOVEMENTS.

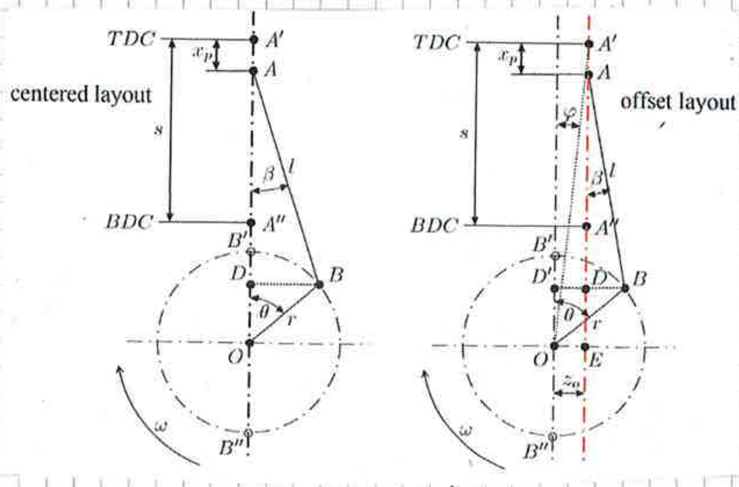


DOWNHILL (INWARD) MOVEMENTS
UPHILL (OUTWARD) MOVEMENTS

- THE WORKING CYCLE CAN BE IMPLEMENTED WITHIN FOUR PISTON STROKES OR TWO PISTON STROKES (STROKE: "CORSA"); THE ENGINE IS CORRESPONDINGLY REFERRED TO AS A: **FOUR-STROKE (4T)** OR A **TWO-STROKE (2T)** ENGINE.

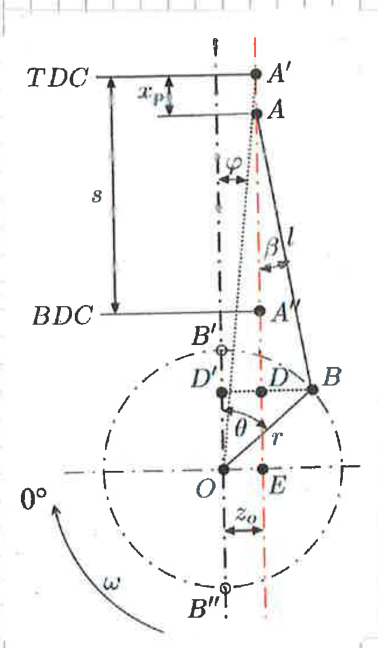
- LAYOUT

- THE **INVESTIGATION** OF THE CRANK MECHANISM **IS DEVELOPED IN TERM OF**:
  → KINEMATICS, IN ORDER TO CALCULATE **DISPLACEMENTS, VELOCITIES** AND **ACCELERATIONS** OF PISTON AND CONNECTING ROD.
  → DYNAMICS, IN ORDER TO EVALUATE **FORCES** AND **MOMENTS** INVOLVED ON THE CRANK MECHANISM, CONNECTING ROD, SINGLE CRANK AND CYL. BLOCK
- THE CRANK MECHANISM KINEMATICS CAN BE OBTAINED **CONSIDERING** SIMPLIFIED SCHEMES WHERE CRANKSHAFT ROTATION AXIS, CRANK PIN AND PISTON POSITIONS ARE REPORTED AS **GEOMETRICAL POINTS** AND THE MAIN DIMENSIONS AND PARAMETERS OF THE CRANK MECHANISM ARE INDICATED.
- THE **LAYOUT** OF THE CRANK MECHANISM CAN BE OF TWO TYPES:
  → CENTERED LAYOUT
  → OFFSET LAYOUT



③

OFFSET CRANK MECHANISM



SYMBOLS ARE THE SAME OF THE CENTERED LAYOUT

E, D' : ADDITIONAL USEFUL CONSTRUCTION POINTS

φ : USEFUL CONSTRUCTION ANGLE $A\hat{O}$

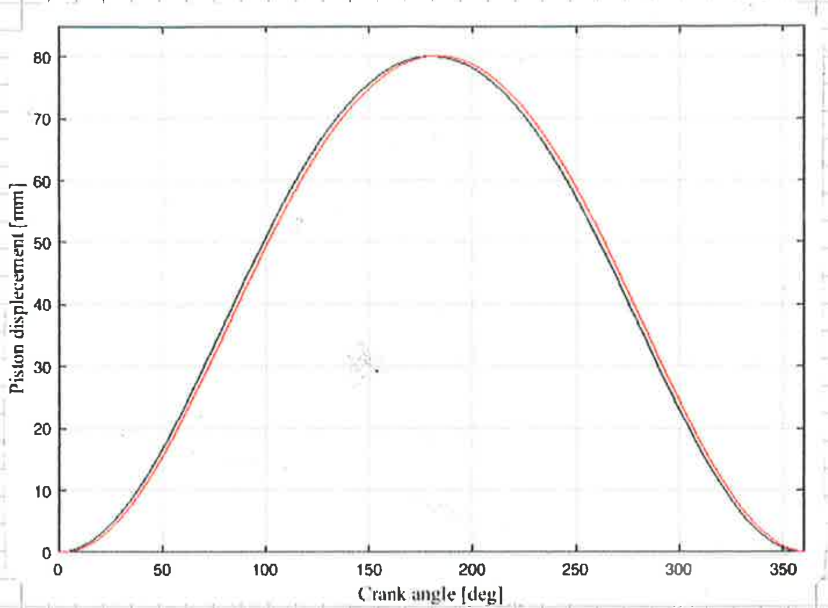
z_o : OVERALL OFFSET

THE OVERALL OFFSET (z_o) IS GIVEN BY THE SUM OF THE DISPLACEMENT OF THE CRANKSHAFT AXIS ($z_{o,c}$) AND THE DISPLACEMENT OF THE WRIST PIN AXIS ($z_{o,wp}$), BOTH CONSIDERED WITH THEIR OWN SIGN (POSITIVE IF IN AGREEMENT WITH THE ROTATION DIRECTION OF THE CRANKSHAFT WHEN THE PISTON REACHES TDC, NEGATIVE OTHERWISE): $z_o = z_{o,c} + z_{o,wp}$.

- K - PISTON DISPLACEMENT, OFFSET VS CENTERED

PISTON DISPLACEMENT OF THE OFFSET (RED) AND CENTERED CRANK MECH. (BLACK):

$R = 40\,[mm]$, $z_0 = 8\,[mm]$, $\omega = 4500\,[Rpm]$, $\Lambda = 0.20$



- K - PISTON VELOCITY, OFFSET

THE PISTON VELOCITY IS NOT UNIFORM AND IT CAN BE EXPRESSED BY THE DERIVATIVE OF THE PISTON DISPLACEMENT WITH RESPECT TO TIME:

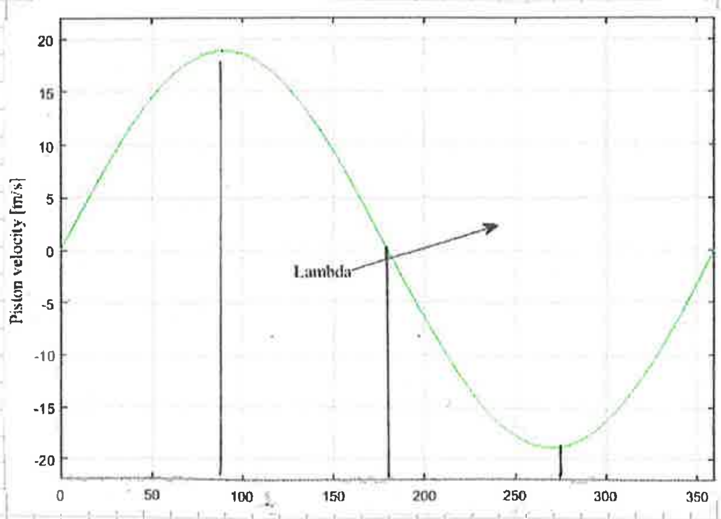
$$V_P = \frac{dx_P}{dt} = \frac{dx_P}{d\theta}\frac{d\theta}{dt} = \frac{dx_P}{d\theta}\,\omega = \omega R\left[\frac{\cos\theta(\Lambda\sin\theta - \delta)}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}} + \sin\theta\right] =$$

$$= \omega R\left[\frac{\Lambda}{2}\,\frac{\sin 2\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}} - \frac{\delta\cos\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}} + \sin\theta\right] =$$

$$= \omega R\left[\sin\theta - \frac{\delta\cos\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}} + \frac{\Lambda}{2}\,\frac{\sin 2\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}}\right]$$

PISTON VELOCITY (V_P) VS CRANK ANGLE (θ):

($R = 40\,[mm]$, $z_0 = 8\,[mm]$, $\omega = 4500\,[Rpm]$, $\Lambda = [0.20, 0.25, 0.30, 0.35]$)



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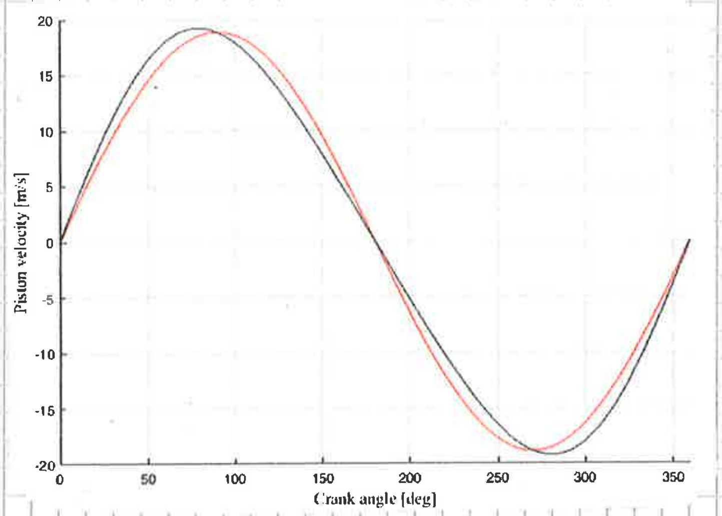
- K - PISTON VELOCITY, OFFSET VS CENTERED (APPROX.)

OFFSET (RED) ; CENTERED (BLACK).

($R = 40$ [mm], $z_0 = 8$ [mm], $\omega = 4500$ [RPM], $\Lambda = 0.20$)



- K - MEAN PISTON VELOCITY, CENTERED

- THE APPROXIMATED INSTANTANEOUS PISTON VELOCITY OF THE CENTERED CRANK MECHANISM IS OFTEN REFERRED TO THE **MEAN PISTON VELOCITY** (u):

$$u = \frac{2 s \cdot n}{60} \quad , \; n \text{ [RPM] ENGINE SPIN SPEED} \qquad \omega = \frac{2\pi n}{60}$$

$$\frac{V_P}{u} = \frac{\pi}{2}\left(\sin\theta + \frac{\Lambda}{2}\frac{\sin 2\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}\right) \approx \frac{\pi}{2}\left(\sin\theta + \frac{\Lambda}{2}\sin 2\theta\right)$$

- THE MEAN PISTON VELOCITY IS A LINEAR VELOCITY AND THEREFORE IT SHOULD BE MEASURED IN [m/s].
- THE MEAN PISTON VELOCITY IS IMPORTANT BECAUSE HIGH STRESSES ARE GENERATED BY THE INERTIA FORCES AND HIGH FLUID LOSSES ARE GENERATED DURING THE REPLACING PROCESS OF THE ENGINE CHARGE.
- FOR COMMERCIAL ENGINES THE MEAN PISTON VELOCITY IS 10÷16 [m/s] WITH CURRENT LIMIT OF ABOUT 22 [m/s] FOR COMPETITION ENGINES.

- K - APPROXIMATED PISTON ACCELERATION, CENTERED

- STARTING FROM THE PISTON ACCELERATION OF THE CENTERED CRANK MECHANISM

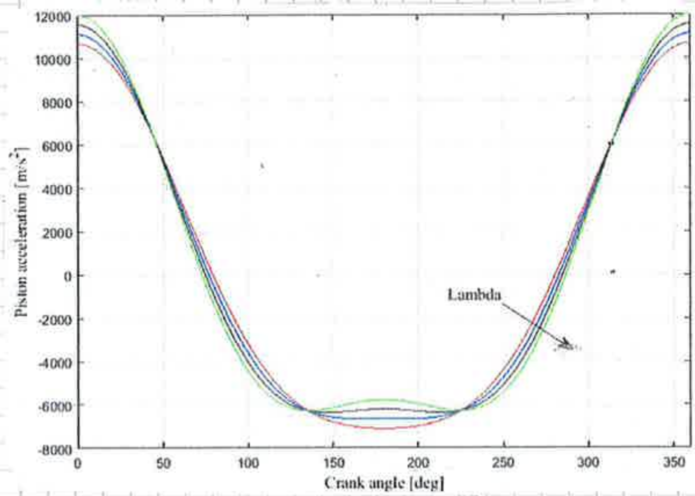
$$a_p = \omega^2 R \left[\cos\theta + \frac{\Lambda \cos 2\theta}{\sqrt{1-\Lambda^2 \sin^2\theta}} + \frac{\Lambda^3}{4} \frac{(\sin 2\theta)^2}{\sqrt{(1-\Lambda^2 \sin^2\theta)^3}} \right]$$

- BY NEGLECTING THE TERM $\Lambda^2 \sin^2\theta$ WITH RESPECT TO THE UNIT, IT REDUCES TO THE SO-CALLED **APPROXIMATED PISTON ACCELERATION**:

$$a_p \approx \omega^2 R (\cos\theta + \Lambda \cos 2\theta)$$

APPROXIMATED PISTON ACCELERATION (a_p) VS CRANK ANGLE (θ):

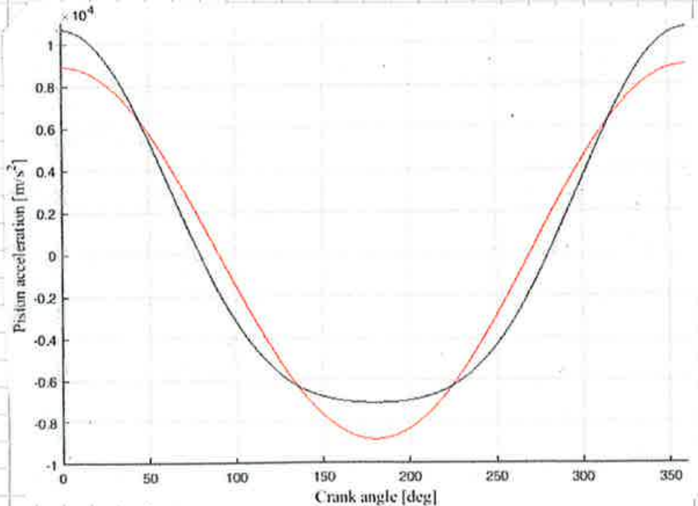
($R = 40$ [mm], $\omega = 4500$ [RPM], $\Lambda = [0.20, 0.25, 0.30, 0.35]$)



- K - PISTON ACCELERATION, OFFSET VS CENTERED (APPROX.)

OFFSET (RED), CENTERED (BLACK)

($R = 40$ [mm], $z_0 = 8$ [mm], $\omega = 4500$ [RPM], $\Lambda = 0.20$)



- K - CONNECTING ROD ANGULAR VELOCITY AND ACCELERATION

- REFERRING TO THE GEOMETRICAL RELATIONS OF THE OFFSET AND CENTERED CRANK MECHANISM:

$\sin\beta = \Lambda \sin\theta - \delta \quad , \quad \sin\beta = \Lambda \sin\theta$

AND DERIVING WITH RESPECT TO TIME:

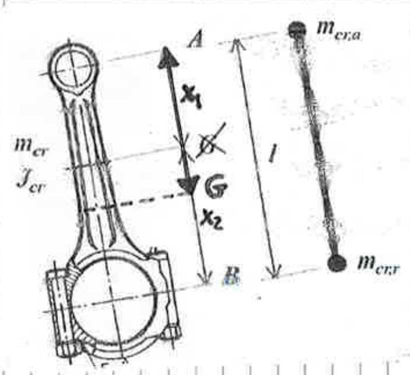
$$\cos\beta \frac{d\beta}{dt} = \Lambda \omega \cos\theta$$

- THE CONNECTING ROD ANGULAR VELOCITY AND ACCELERATION ARE THEN:

$$\omega_{CR} = \dot{\beta} = \frac{d\beta}{dt} = \omega \Lambda \frac{\cos\theta}{\cos\beta} = \omega \Lambda \frac{\cos\theta}{\sqrt{1 - \Lambda^2 \sin^2\theta}}$$

$$\dot{\omega}_{CR} = \ddot{\beta} = -\omega^2 \Lambda \left(\frac{\sin\theta}{\cos\beta} - \Lambda \frac{\sin\beta \cos^2\theta}{\cos^3\beta} \right) = -\omega^2 \Lambda \left[\frac{\sin\theta}{\sqrt{1 - \Lambda^2 \sin^2\theta}} - \Lambda^2 \frac{\sin\theta \cos^2\theta}{\left(\sqrt{1 - \Lambda^2 \sin^2\theta}\right)^3} \right]$$

- THE CONNECTING ROD HAS A ROTO-TRANSLATIONAL MOTION BECAUSE ITS SMALL EYE (OR SMALL END OR FOOT), ATTACHED TO THE PISTON, HAS A TRANSLATIONAL MOTION AND, ON THE CONTRARY, ITS BIG EYE (OR BIG END OR HEAD) HAS A ROTATIONAL MOTION BECAUSE IT IS STRICTLY LINKED TO THE CRANKPIN (A BUSHING BEARING, OR SIMPLY BUSHING, IS PLACED IN BOTH THE SMALL AND THE BIG EYE) (BUSCING: "BOCCOLA").
- THESE TWO CONNECTION POINTS HAVE TO BE CONSIDERED AS THE LOCATION POINTS USUALLY USED TO SIMPLIFY THE CONNECTING ROD INERTIAL PROPERTIES: THE CONNECTING ROD CAN BE THEN REPLACED BY A SYSTEM WITH TWO CONCENTRATED MASSES LINKED BY A THIN BAR WITHOUT MASS.



- MASS $m_{CR,a}$ IS CONCENTRATED IN THE CENTRE OF THE SMALL EYE (FOOT) AND MASS $m_{CR,R}$ IN THE CENTRE OF THE BIG EYE (HEAD).
- THE LUMPED (= CONCENTRATED) PARAMETER SYSTEM IS DYNAMICALLY EQUIVALENT TO THE CONTINUOUS BODY ONLY IF THE CONSERVATION OF TOTAL MASS, CENTRE OF GRAVITY AND TOTAL MOMENT OF INERTIA ARE ENSURED.

- D - CONNECTING ROD REDUCTION

$$\begin{cases} m_{CR} = m_{CR,a} + m_{CR,R} \\ m_{CR,a} x_1 = m_{CR,R} x_2 \\ m_{CR,a} x_1^2 + m_{CR,R} x_2^2 + J_0 = J_{CR} \end{cases} \Rightarrow \begin{cases} m_{CR,a} = m_{CR} \frac{x_2}{l} \\ m_{CR,R} = m_{CR} \frac{x_1}{l} \\ J_0 = J_{CR} - m_{CR} x_1 x_2 \end{cases}$$

- THE ADDITIONAL MOMENT OF INERTIA J_0 IS ALWAYS NEGATIVE (THEN IT HAS NO PHYSICAL MEANING BUT IT IS NECESSARY TO GUARANTEE THE CONSERVATION OF THE TOTAL MOMENT OF INERTIA)
- FOR A FOUR-STROKE (4T) ENGINE THE COMMON AVERAGE VALUES ARE OF ABOUT:

$$\begin{cases} m_{CR,a} = (0.2 \div 0.35)\, m_{CR} \\ m_{CR,R} = (0.8 \div 0.65)\, m_{CR} \\ J_0 = -(0.01 \div 0.03)\, m_{CR} \frac{R^2}{\lambda^2} \end{cases}$$

$m_{CR,a}$ MEANING OF "a"? RECIPROCATING PARTS / PISTON GROUP/ASSEMBLY?

$m_{CR,R}$ MEANING OF "R"? ROTATING PARTS / CRANK GROUP/ASSEMBLY?

→ ALTERNATING

→ ROTATING

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- D - RECIPROCATING INERTIAL FORCE

- THE TOTAL MASS OF RECIPROCATING PARTS (PISTON GROUP) IS:

$m_a = m_p + m_{WP} + m_{CR,a}$

m_p: PISTON MASS (INCLUDING ELASTIC RINGS)
m_{WP}: WRIST PIN MASS
$m_{CR,a}$: CONNECTING ROD MASS CONCENTRATED IN THE SMALL EYE (A).

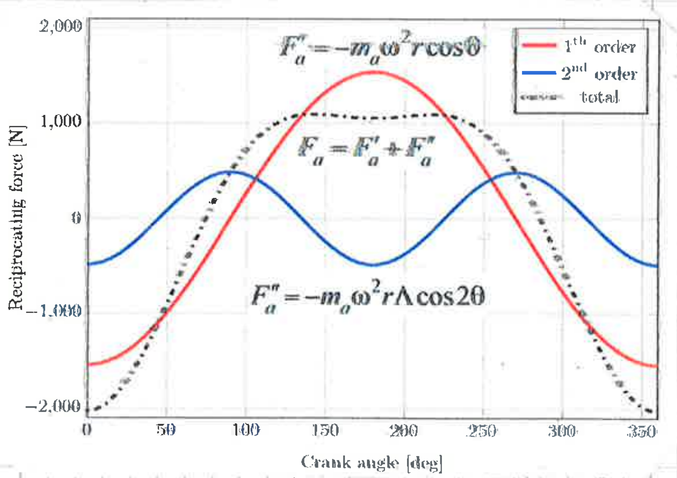
- REFERRING TO THE CENTERED CRANK MECHANISM LAYOUT, AND ASSUMING THE APPROXIMATED EXPRESSION OF THE PISTON ACCELERATION, THE RECIPROCATING PARTS THAT MOVE ALONG THE CYLINDER AXIS ARE THEN SUBJECTED TO THE INERTIAL FORCE:

$F_a = -m_a a_p = -m_a \omega^2 R(\cos\theta + \Lambda \cos 2\theta) = F_a' + F_a''$: RECIPROCATING INERTIAL FORCE

- FORCE F_a THEN CONSISTS OF THE SUM OF TWO TERMS (A FIRST-ORDER AND A SECOND-ORDER TERM) AND IT IS STEADILY DIRECT ALONG THE CYLINDER AXIS:

$F_a' = -m_a \omega^2 R \cos\theta$: FIRST-ORDER TERM / PRIMARY RECIPR. FORCE
$F_a'' = -m_a \omega^2 R \Lambda \cos 2\theta$: SECOND-ORDER TERM (DOUBLE FREQUENCY) / SECONDARY R.F.

TREND OF THE TWO TERMS BESIDE THEIR SUM F_a, WITH RESPECT TO THE CRANK ANGLE:



- THE DIRECTION OF THE INERTIAL FORCE F_a IS FIXED IN TIME WHILE ITS AMPLITUDE AND ORIENTATION ARE FUNCTIONS OF THE CRANK ANGLE (θ) (I.E. OF CRANK ANGULAR POSITION).
- THE AMPLITUDE OF F_a ALSO DEPENDS ON THE ACTUAL ANGULAR SHIFT THAT EXISTS BETWEEN THE CRANKS.
- IN CASE OF V-ENGINES, THE AMPLITUDE OF F_a ALSO DEPENDS ON THE ENGINE BANKS ANGLE.
- F_a IS NOT A ROTATING VECTOR, AS THE CENTRIFUGAL FORCE IS, BUT ITS TERMS CAN BE DECOMPOSED IN THE SUM OF ROTATING VECTORS THAT REPRESENTS THE STARTING POINT FOR BALANCING THE ENGINE RECIPROCATING FORCES AND NEUTRALIZING (COMPLETELY OR PARTIALLY) THE CORRESPONDING ENGINE VIBRATIONS.

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- SHORT STROKE ENGINES (OR OVER-SQUARE ENGINES : D>S) CAN ROTATE AT HIGHER SPIN SPEED AND ARE SUBJECTED TO AVAILABILITY OF LOW RECIPROCATING INERTIAL FORCE.
- THEY ALLOW A CONFORTABLE ACCOMODATION OF VALVES WITH GENEROUS DIMENSIONS, AND THEN ALLOW A BETTER ADMISSION OF CHARGE, FRESH AIR AND FUEL, INTO THE CYLINDER.
- THEY HAVE ALSO A LOWER OVERALL HEIGH WHICH ALLOWS AN EASIER ACCOMODATION IN CARS WITH LOW HOOD AND A BETTER VISIBILITY FROM INSIDE.

- LONG-STROKE ENGINES (OR UNDER-SQUARE ENGINES : D<S): BY INCREASING THE STROKE LENGTH (S) THE PISTON TRANSLATIONAL VELOCITY REDUCES AND THE ENGINE HAS TO OPERATE AT LOW SPIN SPEED, PRIVILEGING HIGH TORQUE, AND CONSEQUENTLY HIGH POWER, AT LOW REGIME (MUST USED IN INDUSTRIAL APPLICATION).
- IN AUTOMOTIVE APPLICATION THE MUST USED STROKE LENGTH IS DEFINED IN ORDER TO OBTAIN A SQUARE ENGINE (I.E. D≈S).

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- D - RESULTANT FORCE ON THE CRANK MECHANISM

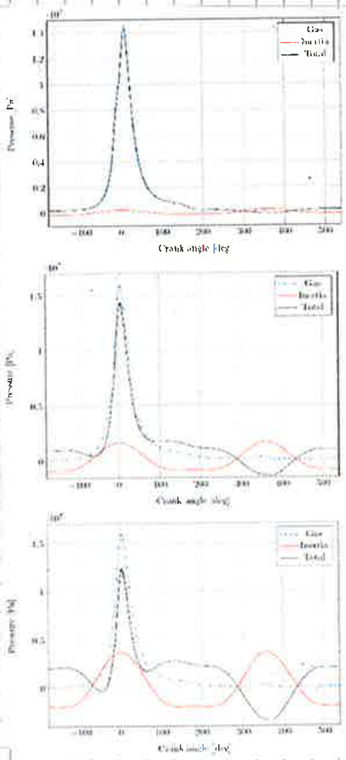
- SUMMARIZING, THE TOTAL MASSES OF RECIPROCATING AND ROTATING PARTS INVOLVED IN THE COMPUTATION OF THE INERTIAL FORCES F_a AND F_R OF THE CRANK MECHANISM ARE:

$m_a = m_p + m_{wp} + m_{cR,a}$: RECIPROCATING MASSES CONCENTRATED AT POINT A

$m_R = m_{cp} + 2 m_{cw} \frac{R_{cw}}{R} + m_{cR,R}$: ROTATING MASSES CONCENTRATED AT POINT B

- THE RESULTANT FORCE $(F = F_g + F_a)$ ACTING ON THE CRANK MECHANISM IS THE COMPOSITION OF THE INSTANTANEOUS VALUES OF THE FORCE DUE TO THE GAS PRESSURE (F_g) AND THE FORCE DUE TO THE RECIPROCATING MASSES (F_a).
- THE RESULTANT FORCE F IS ALWAYS DIRECTED ALONG THE CYLINDER AXIS
- THE TREND OF FORCE F WITH RESPECT TO THE CRANK ANGLE DEPENDS ON THE ROTATION REGIME OF THE ENGINE:
  - AT LOW ENGINE REGIME THE GAS PRESSURE ACTION IS DOMINANT.
  - AT MEDIUM ENGINE REGIME THE INERTIAL ACTIONS BEGIN TO BE CONSIDERABLE.
  - AT HIGH ENGINE REGIME THE INERTIAL ACTIONS BECOME MORE IMPORTANT AND REGULARIZE THE TREND OF THE RESULTANT FORCE (F).

TYPICAL TRENDS OF THE GAS PRESSURE FORCE (F_g) AND OF THE RECIPROCATING INERTIAL FORCE (F_a) (NORMALIZED WITH RESPECT TO THE PISTON CROSS SECTION, I.E. THE EQUIVALENT INERTIAL PRESSURE)

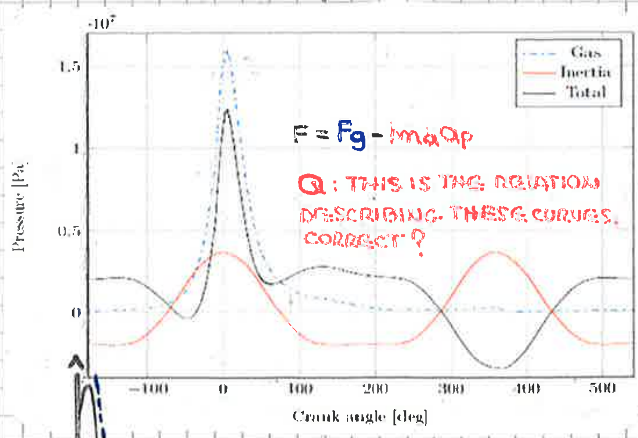


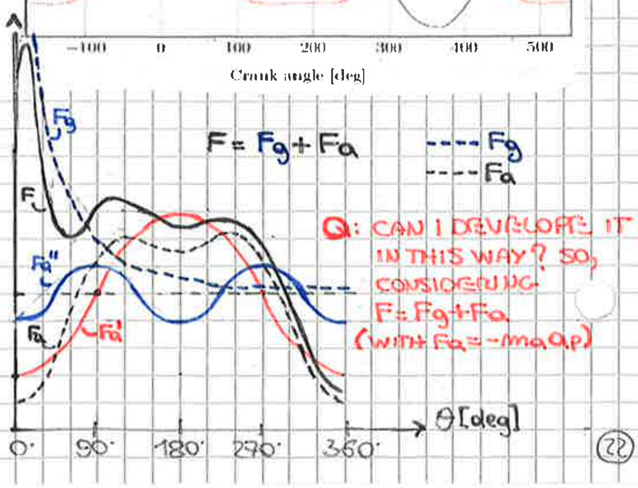
$\omega = 1500$ [RPM]

$\omega = 4000$ [RPM]

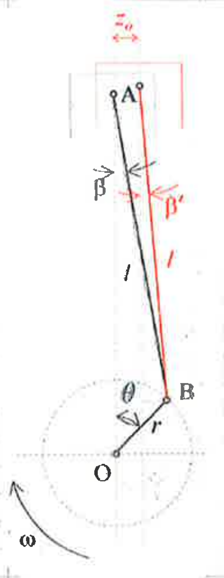
$\omega = 6000$ [RPM]

ZOOM





- AS MENTIONED BEFORE, IN A CRANK MECHANISM **WITH OFFSET LAYOUT** THE FORCE F_m **IS LOWER** THAN IN THE CENTERED LAYOUT
- AS AN EXAMPLE, CONSIDER $\ell = 150$ [mm], $R = 40$ [mm], CRANK ANGLE $\theta = 50° = 0.872$ [Rad] WITH RESPECT TO TDC, PISTON RESULTANT FORCE $F = 6400$ [N] AND OFFSET $z_0 = 12$ [mm]



CENTERED LAYOUT:

$\sin\beta = \frac{R}{\ell}\sin\theta = 0.204 \Rightarrow \beta = 0.20544$ [Rad] $= 11°46'45''$

$F_m = F\tan\beta = 1334$ [N]

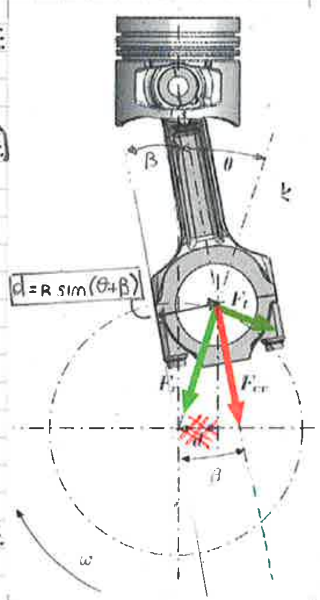
OFFSET LAYOUT:

$\sin\beta' = \frac{R\sin\theta - z_0}{\ell} = 0.124 \Rightarrow \beta' = 0.1245$ [Rad] $= 7°8'4''$

$F_m = F\tan\beta' = 800$ [N] (ALSO $F_{CR}^{OFF} < F_{CR}^{CENT}$)

- D - TORQUE ON THE SINGLE CRANK

- CONSIDERING A **CENTERED** CRANK MECHANISM, THE FORCE F_{CR} ACTING ALONG THE CONNECTING ROD, HAS **ARM d** WITH RESPECT TO THE CRANKSHAFT AXIS; THE TORQUE (CLOCKWISE) ON THE SINGLE CRANK IS:



$$M = F_{CR}\cdot d = F_{CR}\cdot R\sin(\theta+\beta) = \frac{F}{\cos\beta}R\sin(\theta+\beta)$$

- CONSIDERING THAT: $\sin(\theta+\beta) = \sin\theta\cos\beta + \cos\theta\sin\beta$;
$\sin\beta = \Lambda\sin\theta$, $\cos\beta = \sqrt{1-\Lambda^2\sin^2\theta}$, WE CAN RE-WRITE:

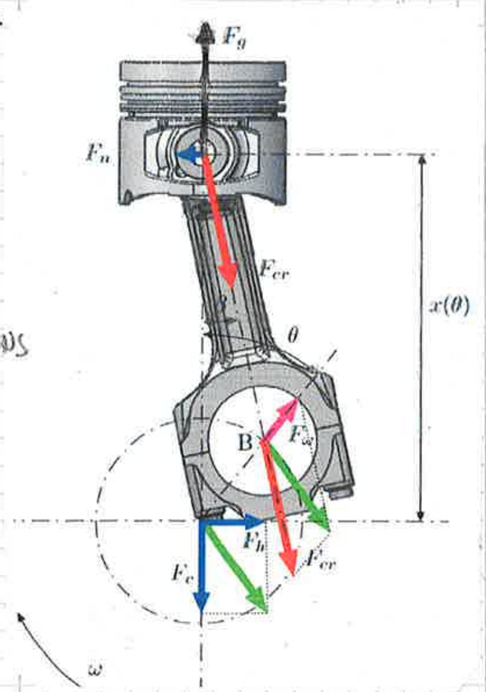
$$M = F\cdot R\left(\sin\theta + \frac{\Lambda\sin\theta\cos\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}\right)$$

- NEGLECTING $\Lambda^2\sin^2\theta$, CONSIDERING $\sin\theta\cos\theta = \frac{1}{2}\sin 2\theta$, THE TORQUE (M) ON THE SINGLE CRANK REDUCES TO:

$$M \approx F\cdot R(\sin\theta + \Lambda\sin\theta\cos\theta) \approx F\cdot R\left(\sin\theta + \frac{\Lambda}{2}\sin 2\theta\right)$$: APPROX. TORQUE ON SINGLE CRANK

- THE **MAXIMUM VALUE** OF THE TORQUE ON THE SINGLE CRANK (CENTERED CRANK MECHANISM) IS **STRONGLY HIGHER** THAN THE **AVERAGE VALUE** WHICH CAN BE COMPUTED AS THE **SUMMATION OF ALL GRAPH AREAS** (POSITIVE AND NEGATIVE) (LOOK AT THE GRAPH →)

- D - FORCES AND MOMENT ON THE CYLINDER BLOCK



- THE GAS PRESSURE INSIDE THE CYLINDER AND THE INERTIAL FORCES DUE TO THE DYNAMICS OF THE INDIVIDUAL COMPONENTS TRANSFER FORCES TO THE ENGINE BLOCK THAT MUST BE COUNTERED BY THE ENGINE MOUNTS.
- AS DONE SO FAR, IT IS CONSIDERED THE CASE OF A SINGLE CYLINDER ENGINE, BUT ALL CONSIDERATIONS MAY BE EXTENDED TO ENGINES WITH MORE CYL.
- AT POINT B THE FORCE F_{CR} HITS THE LINE OF ACTION OF THE FORCE F_ω THAT ACTS ON EVERY ROTATING MASSES.
- THESE TWO FORCES (F_{CR} AND F_ω) GENERATE AN HORIZONTAL FORCE (F_h) AND A VERTICAL FORCE (F_V) ACTING ON THE MAIN BEARINGS AND THEN ON THE CYLINDER BLOCK.

$$\begin{cases} F_h = F_{CR}\sin\beta + F_\omega \sin\theta = F_m + F_\omega \sin\theta \\ F_V = F_{CR}\cos\beta - F_\omega\cos\theta = F - F_\omega\cos\theta = F_g + F_a - F_\omega\cos\theta \end{cases}$$: FORCES ON THE MAIN BEARINGS

- THE SINGLE CRANK IS FINALLY LOADED BY AN HORIZONTAL FORCE WITH THE SAME DIRECTION OF F_h, BY A VERTICAL FORCE WITH OPPOSITE DIRECTION WITH RESPECT TO F_V, AND BY A COUNTERCLOCKWISE EQUILIBRIUM MOMENT.

$$\begin{cases} F_H = F_h - F_m = F_\omega \sin\theta \\ F_V = F_V - F_g = F_a - F_\omega\cos\theta \end{cases}$$: OVERALL FORCES ACTING ON THE CYLINDER BLOCK

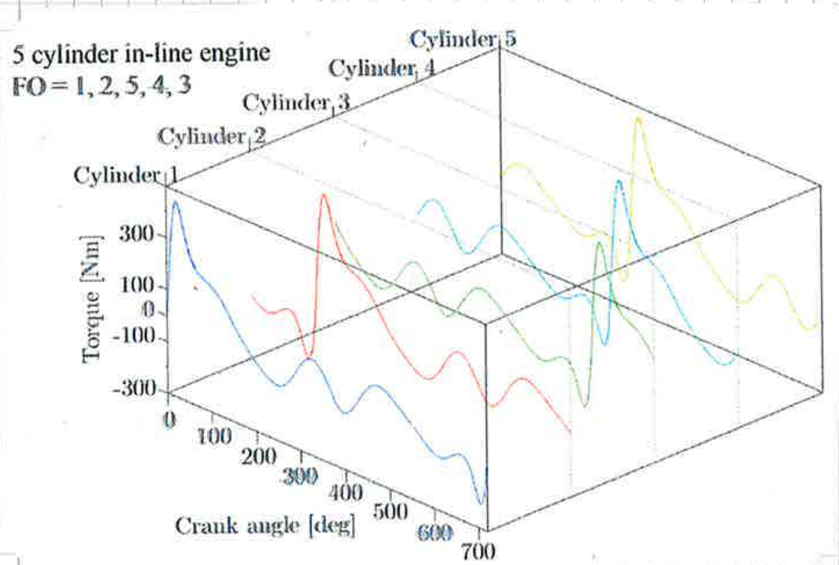
$$M_e = F_m\, x(\theta) = F_m(\ell\cos\beta + R\cos\theta)$$: COUNTERCLOCKWISE EQUILIBRIUM MOMENT ON C.B.

- THE VERTICAL FORCE F_V DOES NOT DEPEND ON THE GAS PRESSURE BECAUSE THE TWO FORCES F_g (ONE ACTING ON THE CYLINDER HEAD, THE OTHER ONE ON THE PISTON) HAVE EQUAL VALUE BUT OPPOSITE DIRECTION, CANCELLING EACH OTHER.
- FURTHERMORE F_V DEPENDS BOTH ON RECIPROCATING AND CENTRIFUGAL FORCES, WHILE F_H DEPENDS ONLY ON CENTRIFUGAL FORCE.

- **MULTI-CYLINDER ENGINE AND MOST STRESSED CRANK**

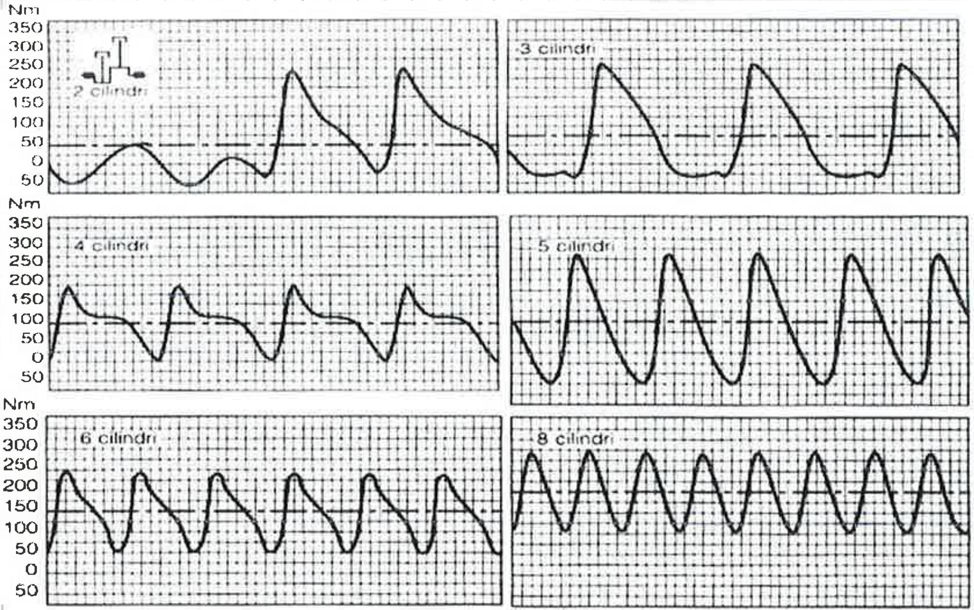
- THE FORCE ANALYSIS DEVELOPED UNTIL NOW IS BASED ON A SINGLE-CYLINDER ENGINE, BUT IT CAN BE APPLIED ON EACH CYLINDER OF A MULTI-CYLINDER ENGINE (BOTH FOR IN-LINE AND V CONFIGURATION)
- IN MULTI-CYLINDER ENGINES, THE TREND OF EACH FORCE AND MOMENT COMPUTED FOR THE SINGLE-CYLINDER ENGINE HAS TO BE SUPERIMPOSED BY RESPECTING THE ENGINE FIRING ORDER (FO) TO OBTAIN THE CORRECT TIME AND SPACE "POSITIONING".
- IN MULTI-CYLINDER ENGINES, THE STRESS ON A SPECIFIC CRANK IS NOT SOLELY DUE BY THE STRESS DUE TO THE PISTON CONNECTED WITH THAT SPECIFIC CRANK BUT ALSO BY THE PRESENCE OF THE OTHER CRANKS.
- TO IDENTIFY THE MOST STRESSED CRANK, THE ENGINE TORQUE ACTING ON EACH "SINGLE-CYLINDER" HAS TO BE COMPUTED, AND THE TRENDS OF ALL CYLINDERS HAVE TO BE ADDED STARTING FROM THE FIRST CYLINDER UNTIL THE LAST ONE INDICATED BY THE FO.

- EXAMPLE: MULTI-CYLINDER ENGINE:



- ENGINE DEGREE OF IRREGULARITY

- THE ENGINE DEGREE OF IRREGULARITY CAN BE REPRESENTED AS THE RATIO BETWEEN THE MAXIMUM AND THE AVERAGE ENGINE TORQUE.
- AS THE NUMBER OF CYLINDERS INCREASES, THE IRREGULARITY DEGREE DECREASES.



Irregularity degree:

2 cyl 4.45
3 cyl 3.47
4 cyl 2.95
5 cyl 2.33
6 cyl 1.65
8 cyl 1.49

- THE IRREGULARITY DEGREE CAN BE EXPRESSED IN TERMS OF KINEMATIC DEGREE OF IRREGULARITY Δ THAT IS DEFINED AS THE RATIO BETWEEN THE DIFFERENCE OF THE MAXIMUM AND MINIMUM INSTANTANEOUS SPEEDS OF ROTATION AND THE NOMINAL SPIN SPEED (ω) OF THE ENGINE:

$$\Delta = \frac{\Omega_{MAX} - \Omega_{min}}{\omega} \quad \text{KINEMATIC DEGREE OF IRREGULARITY}$$

- AS THE INSTANTANEOUS SPIN SPEED (ω) IS QUITE EQUAL TO THE MEAN SPIN SPEED

$$\omega \approx \omega_m = \frac{\Omega_{MAX} + \Omega_{min}}{2}$$

THEREFORE:

$$\Delta = \frac{\Omega_{MAX} - \Omega_{min}}{\omega}\,\frac{2\omega}{2\omega} \approx \frac{\Omega_{MAX} - \Omega_{min}}{\omega}\,\frac{2\omega_m}{2\omega} = \frac{(\Omega_{MAX} - \Omega_{min})\, 2\left(\frac{\Omega_{MAX} + \Omega_{min}}{2}\right)}{2\omega^2} =$$

$$= \frac{\Omega_{MAX}^2 - \Omega_{min}^2}{2\omega^2}$$

- BY INTRODUCING THE:

$$\Delta E = \frac{1}{2} J_e \left(\Omega_{MAX}^2 - \Omega_{min}^2\right) \quad \text{MAXIMUM VARIATION OF KINETIC ENERGY}$$

J_e: DYNAMIC MOMENT OF INERTIA OF THE ENGINE FLYWHEEL (= "VOLANO")

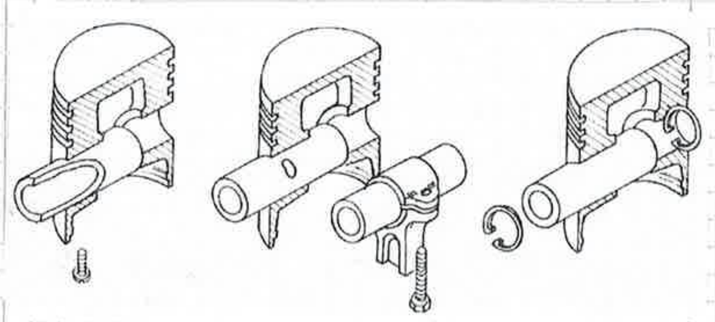
THEREFORE:

$$\Delta = \frac{\Omega_{MAX} - \Omega_{min}}{\omega} = \frac{\Omega_{MAX}^2 - \Omega_{min}^2}{2\omega^2} = \frac{2\Delta E}{J_e 2\omega^2} = \frac{\Delta E}{J_e \omega^2} \quad \text{KINEMATIC DEGREE OF IRREGULARITY}$$

N.W. WE CANNOT DIMENSION THE FLYWHEEL IN A SINGLE OPERATION;
WE START WITH A FIRST DIMENSION PROCEDURE FOR THE FLYWHEEL,
THEN WE DEVELOPE THE ANALYSIS OF THE OTHER COMPONENTS, IN PARTICULAR OF CR.SHAFT
WHITHIN THE CRANKSHAFT ANALYSIS WE ARE ABLE TO IDENTIFY THE LEVEL OF TORSIONAL VIBRATIONS; IF THE TORSIONAL VIBRATIONS CORRESPOND TO A SPIN SPEED (ω) THAT IS DIFFERENT FROM WHICH WE ASSUME TO DESIGN THE SIZE OF THE FLYWHEEL WE HAVE TO COME BACK, CHANGE THIS VELOCITY ($\omega \to \omega'$) AND RE-SIZE THE FLYWHEEL
WE NEED A CERTAIN NUMBER OF ITERATIONS.

N.W. THE FLYWHEEL IS A MECHANICAL COMPONENT USED TO MAINTAIN THE ANGULAR VELOCITY OF THE CRANKSHAFT CONSTANT LIMITING THE EXCESSES OF MECH. E. ON THE TOTAL MECH. WORK (AND VICEVERSA).

• **GEOMETRY AND MATERIALS**



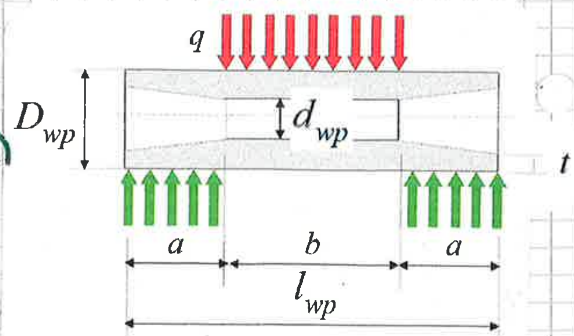
NOTE: "SMALL END" BETTER THAN "EYE"

- THE **WRIST PIN** (OR SIMPLY PIN) IS THE **CONNECTING ELEMENT** BETWEEN THE PISTON AND THE CONNECTING ROD.
- **TO REDUCE** THE **RECIPROCATING MASSES**, ITS GEOMETRY IS AN **HOLLOW CYLINDER** WITH **TAPERED ENDS**, AND **THICKER IN THE MIDDLE**.
- IT IS INSERTED INTO THE **PISTON HUBS** (="MOZZI"), IT PASSES THROUGH THE **SMALL EYE** OF THE CONNECTING ROD AND IT IS **KEPT IN POSITION** BY USING **SNAP ELASTIC RINGS** (="ANELLI ELASTICI A SCATTO") INSERTED INTO THE **GROOVES OF THE PISTON HUBS** (RELATIVE MOTION OCCURS BETWEEN PIN AND SMALL EYE AND BETWEEN PIN AND PISTON) **OR** BY USING A **SCREW** (RELATIVE MOTION OCCURS ONLY BETWEEN PIN AND SMALL EYE).

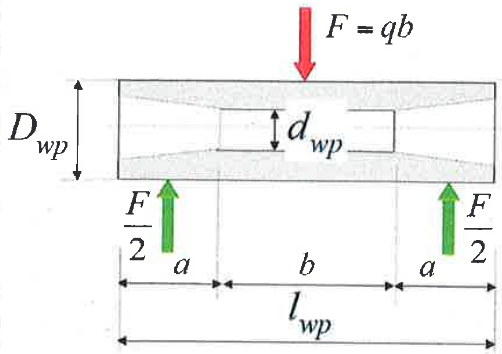
- TWO DIFFERENT CONFIGURATIONS CAN BE ADOPTED:
  - **FIXED PIN** (OR **PIN-DRIVE**) **LOW STRESSED ENGINES**
  - **FLOATING PIN**, **MEDIUM** AND **HIGH STRESSED ENGINES**

- **IN BOTH CONFIGURATION** THE **PIN** IS **FREE TO ROTATE** WITH RESPECT TO THE **CONNECTING ROD**, BY MEANS OF **BUSHING** OR **NEEDLE BEARING**;
- IN THE **FLOATING PIN CONFIGURATION** **RELATIVE MOTION OF THE PIN IS SHARED** BETWEEN CONNECTING ROD AND PISTON AND THEN THE **STRESS PEAKS** ARE **SHARED ON TWO CONTACT SURFACES**.

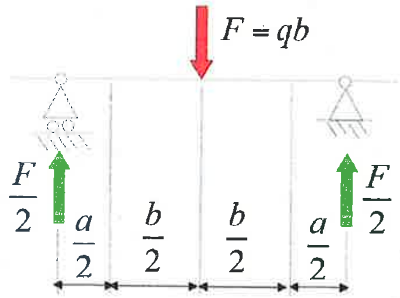
- **DESIGN GUIDELINES**



- THE PIN IS MODELED AS A SUPPORTED BEAM LOADED BY THREE DISTRIBUTED FORCES. (EXTERNAL LOAD IN RED, CONSTRAINT REACTIONS IN G.)
- THE ACTUAL FORCES ARE DISTRIBUTED WITH TRAPEZOIDAL DISTRIBUTION BUT THEY CAN BE SIMPLIFIED ASSUMING UNIFORM DISTRIBUTION.



- THE COMPUTATION MODEL IS A BEAM SUPPORTED AT THE ENDS BY ONE HINGE ( ) AND ONE ROLLER SUPPORT ( ) (BOTH POSITIONED IN THE MIDSPAN SECTION OF THE WRIST PIN SURFACES THAT ARE COUPLED WITH THE PISTON HUBS) AND LOADED BY THREE CONCENTRATED FORCES EQUIVALENT TO THE UNIFORM DISTRIBUTED ONES.



- THE VALUE OF THE EXTERNAL LOAD (F) DEPENDS ON THE TYPE OF CALCULATION:
  - STATIC (INITIAL CONDITION, MAXIMUM TORQUE REGIME): MAXIMUM COMPRESSIVE FORCE, DUE TO THE GASES INERTIAL FORCES ARE NEGLECTED. $F = F_{g,max} = p_{g\,max} \frac{\pi D^2}{4}$
  - FATIGUE (TDC, MAXIMUM SPIN SPEED REGIME): FATIGUE CYCLE BETWEEN $F_{max} = F_{g,max}$ AND $F_{min} = F_{a,max}$

- WRIST PIN LENGTH IS ASSUMED EQUAL TO THE BORE: $l_{wp} = D$
- AS FIRST ATTEMPT, PIN OUTER DIAMETER IS ASSUMED EQUAL TO 35% OF THE BORE, INNER DIAMETER IS 40% OF THE OUTER DIAMETER, AND MINIMUM THICKNESS IS 50% OF THE INNER DIAMETER ($l_{wp} = D$, $D_{wp} = 0.35D$, $d_{wp} = 0.40 D_{wp}$, $t = 0.50 d_{wp} = \frac{d_{wp}}{2}$)

- THE PIN LENGTHS **a** AND **b** CAN BE COMPUTED BY IMPOSING, AS A FIRST APPROXIMATION, AN ADMISSIBLE CONTACT PRESSURE OF 65 [MPa] ON THE PISTON (AND ON THE BUSHING, IN CASE OF FLOATING PIN CONFIGURATION).

$$\begin{cases} \frac{F}{A_{BUSHING}} = \frac{qb}{b D_{wp}} = \frac{q}{0.35D} \le p_{ADM,\,BUSHING} = 65\ [MPa] \\ \frac{F/2}{A_{HUB}} = \frac{qb}{2a D_{wp}} = \frac{q \cdot b}{2a\, 0.35D} = \frac{q \cdot b}{a\, 0.7D} \le p_{ADM,\,PISTON} = 65\ [MPa] \\ (l_{wp} = D = 2a + b) \end{cases} \Rightarrow a, b$$

(FIRST ATTEMPT OF DESIGN DEFINITION)

- THE VON MISES EQUIVALENT STRESS IN THE MIDSPAN SECTION OF THE WRIST PIN, IS COMPUTED COMBINING THE MAXIMUM OVALIZATION STRESS AND THE SHEAR STRESS:

$$\sigma_{EQ} = \sqrt{\sigma_{ov(0,\pi)}^2 + 3\tau^2}$$

- TO SET THE FINAL INNER AND OUTER DIAMETERS OF THE WRIST PIN, THE CHOSEN PARAMETER IS THE MAXIMUM VALUE BETWEEN THE EQUIVALENT STRESS (σ_{EQ}) AND THE PREVIOUS BENDING STRESS (σ_b), THAT HAS TO BE LOWER WITH RESPECT TO THE ADMISSIBLE STRESS ($\sigma_{ADM,WP}$) OF THE WRIST PIN MATERIAL:

$$MAX(\sigma_{EQ}, \sigma_b) \leq \sigma_{ADM,WP} \Rightarrow \text{FINAL } D_{WP}, d_{WP}$$

- ANOTHER VERIFICATION OF THE WRIST PIN IS RELATED WITH ITS LONGITUDINAL AND TRASVERSAL DEFORMATION THAT CAN CAUSE PROBLEMS TO THE LUBRICANT OIL FILM, COMPROMISING THE CORRECT OPERATING CONDITION OF THE GROUP WRIST PIN - CONNECTING ROD. (IT IS ALWAYS A STATIC VERIFICATION)
- THE WRIST PIN MAXIMUM LONGITUDINAL DISPLACEMENT (f) CAN BE COMPUTED BY USING THE STRUCTURAL MECHANICS THEORY AS:

$$f = \left(1 - \frac{b}{2(a+b)}\right)\frac{(a+b)^3 F}{48 E I_{WP}} \quad \left(= v(x)_{MAX} \; ; \; \frac{d^2v}{dx^2} = \frac{M(x)}{E I_{WP}} \; ; \; f v(x) \rightarrow x\right)$$

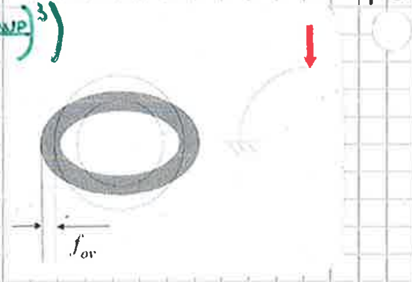
- THE LONGITUDINAL DISPLACEMENT MUST BE LOWER THAN AN ADMISSIBLE VALUE DERIVED FROM DESIGNER EXPERIENCE OR COMPUTATIONAL FLUID-DYNAMICS (CFD) ANALYSIS OF THE LUBRICANT OIL FILM:

$$f \leq f_{ADM} \quad (\text{WITH CFD ANALYSIS OR EXP. INVEST.} \rightarrow f_{ADM})$$

- THE WRIST PIN MAXIMUM TRANSVERSAL DISPLACEMENT (f_{ov}) IS GIVEN BY:

$$f_{ov} = \frac{32 F R_{WP}^3}{e_{WP}(D_{WP} - d_{WP})^3 E} \quad \left(f_{ov} = \frac{F R_{WP}^3}{3 E I_{WP}} \; ; \; I_{WP} = \frac{e_{WP} \cdot h^3}{12} = \frac{e_{WP}}{12}\left(\frac{D_{WP} - d_{WP}}{2}\right)^3\right)$$

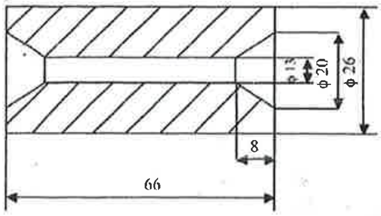
$F = q \cdot b$



- THE TRANSVERSAL DISPLACEMENT MUST BE LOWER THAN THE MINIMUM TOLERANCE (δ_{min}) ASSUMED FOR THE BUSHING (WHICH DEPENDS ON THE BEHAVIOUR OF THE LUBRICATION OIL FILM):

$\delta_{e} \leq \delta_{0m}$ (WITH CFD ANALYSIS OR EXP. INVEST. $\rightarrow \delta_{min}$)   (WE ARE ANALYZING DEF. IN THE FRONTAL PLANE)

• EXAMPLE : WRIST PIN STATIC AND FATIGUE ANALYSIS



DATA

WRIST PIN MATERIAL : 34NiCrMo16

UTS (R_m) = 1420 [MPa] (ULTIMATE TENSILE STRENGTH)

$R_{p0.2}$ = 1030 [MPa]

σ_{D-1} = 700 [MPa]

D (BORE) = 82 [mm] $\quad \ell_{wp} = D = 82$ [mm] ; $D_{wp} = 0.35 D = 28.7$ [mm] ; $d_{wp} = 0.40 D_{wp} = 11.48$ [mm]

S (STROKE) = 90.4 [mm] $\quad \bar{R}_{wp} = \frac{d_{wp}}{2} + \left(\frac{D_{wp} - d_{wp}}{4}\right) = 10.045$ [mm] ; $t_{min} = 0.5\, d_{wp} = 5.74$ [mm]

CONROD LENGTH = 145 [mm] $\quad \frac{(D_{wp} - d_{wp})}{2} = 8.1$ [mm] $> t_{min}$ ✓

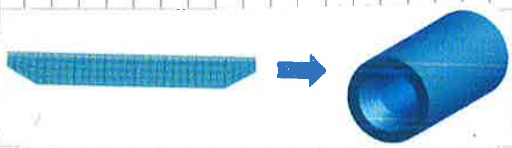
Λ (ELONGATION RATIO) = 0.312

P_g (WORKING PRESSURE) = 140 [bar] ≅ 14 [MPa]

m_a (PISTON + RINGS MASS) = 0.560 [Kg]

m, ω (MAXIMUM SPIN SPEED) = 5500 [Rpm] ≙ 576 [rad/s]

- **2D PRE-MESH = QUAD4 FE** (QUADRATIC SHAPE FUNCTION) (THERE IS A COMMAND TO PASS FROM 2D → 3D)
- **3D MESH = HEXA FE** (SWEEP AND REVOLUTION, EQUIVALENCE)

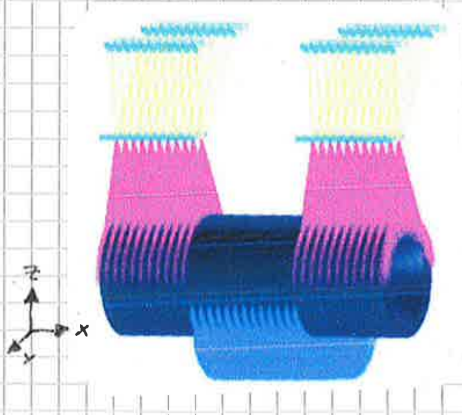


- **CONSTRAINTS :**

– **SPRING SYSTEM (STIFFNESS , 80 [KN/mm]) ALONG Y AND Z** DIRECTION **TO SIMULATE THE PISTON HUBS COMPLIANCE.**

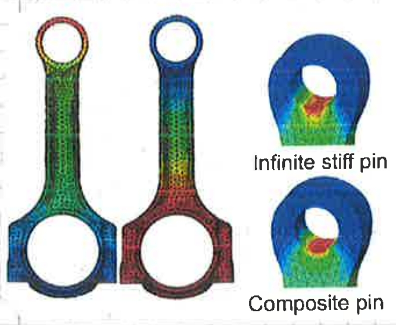
– **ADDITIONAL NODES** (CYAN) **SIMULATE THE HUBS**, THEY ARE CONNECTED WITH THE RESPECTIVE NODES OF THE WRIST PIN MESH BY MEANS OF **MULTI POINT CONSTRAIN ELEMENTS** (MPC, PINK) TO CREATE THE LINK BETWEEN THE SPRINGS AND THE PISTON HUBS.

– **RBE2 ELEMENTS** (YELLOW) **REPRESENT THE SPRINGS** ON Y AND Z DIRECTION.



40

E3 CONNECTING ROD



- INTRODUCTION
- ARCHITECTURE AND GEOMETRY
  - A & G - I CROSS SECTION
  - A & G - H CROSS SECTION
  - A & G - FACE JUNCTION
  - A & G - SMALL EYE
  - A & G - BIG EYE
- MATERIALS AND TREATMENTS
- LOAD CONDITIONS
  - LC - TENSILE LOAD
  - LC - BUCKING LOAD
  - LC - BENDING LOAD
- DESIGN GUIDELINES
  - DG - STEM
  - DG - SMALL EYE
- VERIFICATION GUIDELINES
  - VG - STEM
  - VG - SMALL EYE
  - VG - BIG EYE, CAP
  - VG - BIG EYE, HEAD

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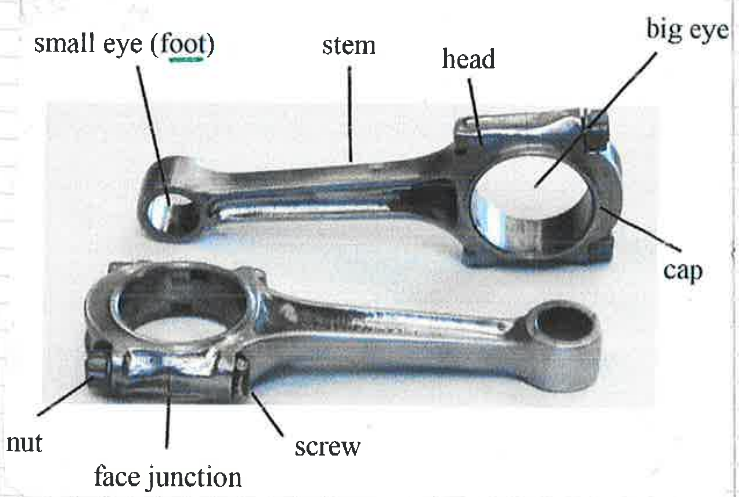

STEM LAYOUTS:

- THE POSSIBLE LAYOUT OF THE STEAM CAN BE:
  - I-ROD STEM: STEM FACE PLACED IN THE PLANE OF RECIPROCATING MOTION (CONNECTING ROD WITH I CROSS SECTION)
  - H-ROD STEM: STEM FACE PLACED IN THE PLANE ⊥ TO RECIPROCATING MOTION (CONNECTING ROD WITH H CROSS SECTION)
- BOTH PROFILES PROVIDE THE REQUIRED STIFFNESS TO THE STEM BUT THEY PRESENT DIFFERENT STRESS DISTRIBUTIONS AND DIFFERENT CONSTRUCTIVE PROBLEMS.

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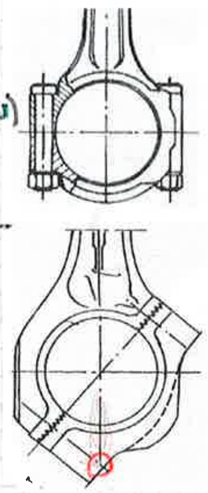
- A & G - FACE JUNCTION



- THE PLANE OF SEPARATION BETWEEN THE HEAD AND THE CAP IS GENERALLY PLACED AT 90° WITH RESPECT TO THE LONGITUDINAL AXIS OF THE STEM (ORTHOGONAL FACE JUNCTION)
- TO REDUCE THE TRANSVERSAL DIMENSION OF THE BIG EYE AND TO FACILITATE THE CROSS-MOUNTING OF THE GROUP PISTON-PIN-CONNECTING ROD THROUGH THE LINER, A CAP WITH AN ANGLE OF 30°-45° IS USED (INCLINED FACE JUNCTION).
- IN THIS CASE THE MAXIMUM STRESS IS CONCENTRATED ON THE CAP WHERE THE CONNECTION FOR THE SCREWS HOUSING (I.E. THE SCREW AXIS) INTERSECTS THE LONGITUDINAL AXIS OF THE CONNECTING ROD.

FIAT SDE 1.3

Isuzu V6 CDTI

- IN ORDER TO AVOID SHEAR STRESS ON THE STEM OF THE SCREWS, THE RELATIVE SLIPPAGE BETWEEN THE CAP AND THE HEAD OF THE BIG EYE IS PREVENTED BY CENTERING THE TWO ELEMENTS WITH DIFFERENT METHODS:
  - SCREWS WITH STEM DIAMETER CALIBRATED IN CORRESPONDENCE TO THE FACE JUNCTION (BOTH FOR ORTHOGONAL AND INCLINED FACE JUNCTION).
  - DOWELS (= "TASSELLI") WITH GRAINS OF 3-5 [mm] DIAMETER. (BOTH FOR ORTHOGONAL AND INCLINED FACE JUNCTION).
  - SERRATION OR TOOTHING ON THE PARTS IN CONTACT. (ONLY FOR INCLINED FACE JUNCTION)

FIAT 1.9 JTD 16V

Honda 2.2 i-CTDi

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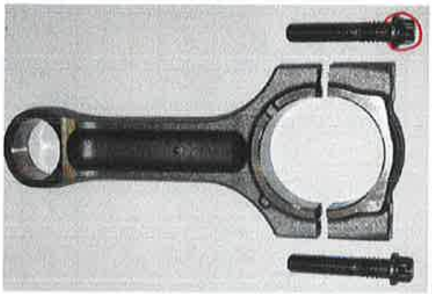
- A & G - BIG EYE

- TO FASTEN ("FISSARE") THE CAP TO THE HEAD OF THE BIG EYE, THE CONNECTING RODS OF MORE TRADITIONAL DESIGN USE BOLTS OF THROUGH TYPE.

Isuzu 1.7 CDTI
(conventional bolts)

- THIS SOLUTION, UNLESS TAKING SPECIAL CARE DURING THE DESIGN, MAY CAUSE WEAKNESS IN THE BIG EYE AT THE BOLT SEAT OF THE HEAD.
- A BETTER SOLUTION IS OBTAINED WHEN A COUNTERSINK SEAT HEAD IS USED WITH RESPECT TO A BROACHING PASS TYPE.
- IN MOST ENGINES THE CAP IS FIXED BY USING SCREWS.

BMW 120d
(screws)



- THIS SOLUTION IS BETTER THAN THE PREVIOUS ONE ONLY IF A PROPER EXECUTION OF THE AXIS POSITION OF THE THREAD IN THE BIG EYE IS GUARANTEED.

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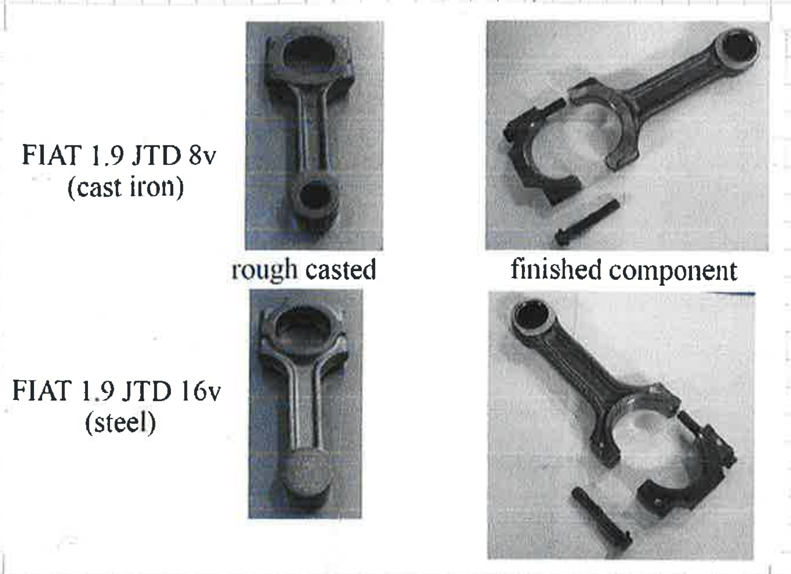
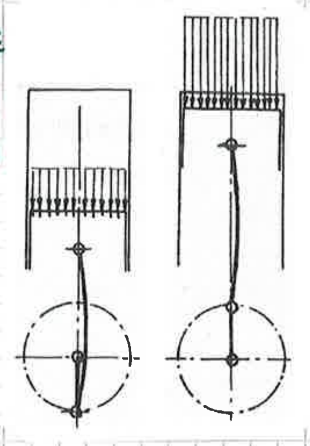
- ON THE FINISHED CONNECTING ROD A SHOT-PEEN TREATMENT IS USUALLY DONE TO INDUCE COMPRESSIVE RESIDUAL STRESSES AND IMPROVE FATIGUE RESISTANCE (UP TO 25%).

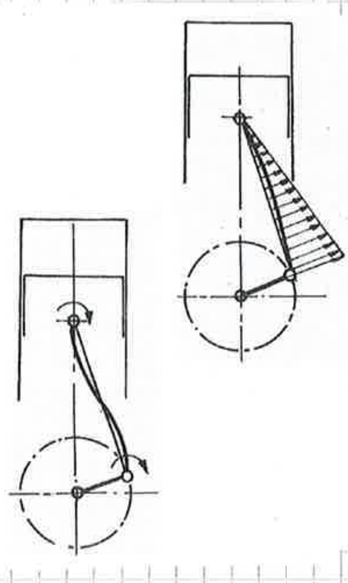


- LC - BUCKLING LOAD



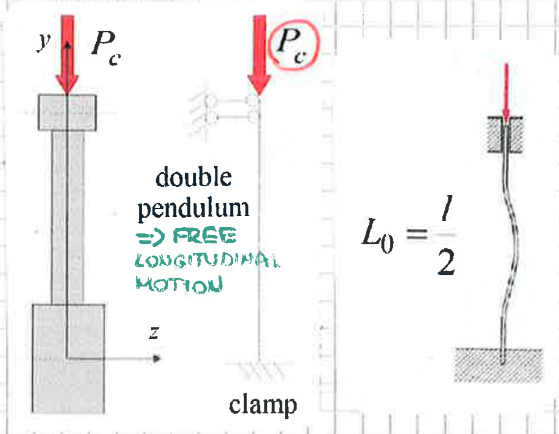
- THE BUCKLING LOAD IS DUE TO THE COMPRESSIVE FORCE GENERATED BY THE GAS EXPANSION WHEN THE PISTON IS AT TDC OR BY THE INERTIAL FORCE WHEN THE PISTON IS AT BDC.
- THE MAXIMUM OF THOSE TWO COMPRESSING FORCES, IN THE WROST CASES RESPECTIVELY OF STARTING OR MAXIMUM SPEED, IS THE INPUT FOR THE BUCKING VALIDATION.

- LC - BENDING LOAD

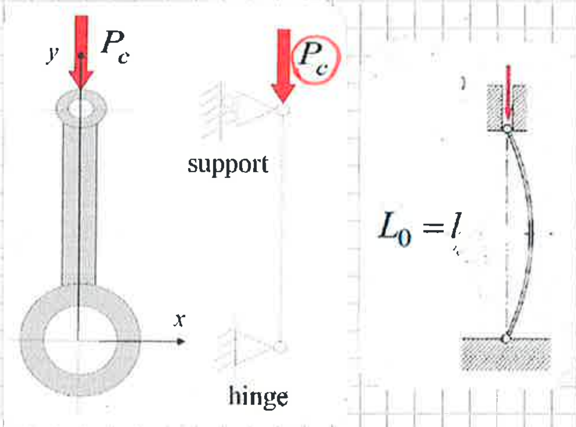


- THE MAIN BENDING LOAD IS DUE TO THE WHIPLASH RESULTING FROM THE ACCELERATION DISTRIBUTION THAT THE CONNECTING ROD UNDERGOES DURING ITS ROTO-TRANSLATIONAL MOTION.
- AN ADDITIONAL BENDING LOAD CAN BE DUE TO THE SEIZURE OF THE BUSHING IN THE SMALL EYE AND/OR IN THE BIG EYE.

- THE BUCKLING MODEL IN THE LATERAL PLANE (ZY) CAN BE SKETCHED AS A BEAM CLAMPED AT ONE END AND WITH FREE AXIAL DEGREE OF FREEDOM AT THE OTHER END. (CLAMP AND DOUBLE PENDULUM)
- THE FREE LENGTH (L_0) OF THIS BEAM IS EQUAL TO THE HALF DISTANCE BETWEEN THE CENTER OF THE SMALL EYE AND THE CENTER OF THE BIG EYE, THAT IS THE HALF-LENGTH $\ell/2$ OF THE CONNECTING ROD. ($L_0 = \ell/2 \Rightarrow \ell = 2 \cdot L_0$)



- THE BUCKLING MODEL IN THE FRONTAL PLANE (XY) CAN BE SKETCHED AS A BEAM SUPPORTED AT ONE END AND CONNECTED TO A HINGE ON THE OTHER END.
- THE FREE LENGTH (L_0) OF THIS BEAM MODEL IS EQUAL TO THE DISTANCE BETWEEN THE CENTER OF THE SMALL EYE AND THE CENTER OF THE BIG EYE, THAT IS THE LENGTH (ℓ) OF THE CONNECTING ROD.



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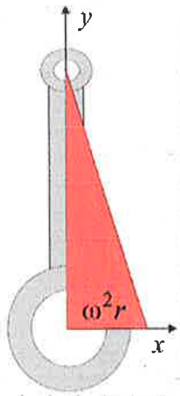
• VERIFICATION GUIDELINES (STEM, SMALL EYE, BIG EYE)

- VG - STEM

• THE BENDING OF THE STEM IN ITS FRONTAL PLANE (XY) DUE TO THE WHIPLASH RESULTS FROM THE ACCELERATION DISTRIBUTION THAT THE CONNECTING ROD STEM UNDERGOES DURING ITS ROTO-TRANSLATIONAL MOTION.

• THE ANGULAR ACCELERATION OF THE CONNECTING ROD HAS A TRIANGULAR DISTRIBUTION WITH NULL VALUE (0) AT THE CENTRE OF THE SMALL EYE AND MAXIMUM VALUE ($\omega^2 R$) AT THE CENTRE OF THE BIG EYE.



• FROM THE CRANK MECHANISM ANALYSIS THE MAXIMUM ANGULAR ACCELERATION OF THE CONN. ROD ($\dot{\omega}_{CR,MAX}$), COMPUTED WITH CRANK ANGLE $\theta = 90°$, IS:

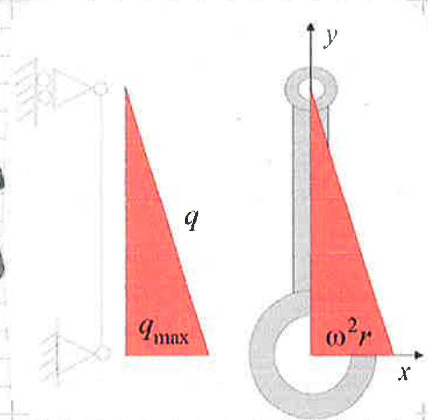
$$\dot{\omega}_{CR,MAX} = \ddot{\beta}_{MAX} = \frac{\omega^2 \Lambda}{\sqrt{1-\Lambda^2}}$$: MAXIMUM ANGULAR ACC. OF C.R. ($\theta = 90°$)

• THE FORCE ACTING ON THE STEM CAN BE COMPUTED AS:

$$\int_V \ddot{\beta}_{MAX}\, dm_{CR} = \int_0^{\ell} \frac{\omega^2 \Lambda}{\sqrt{1-\Lambda^2}} \rho A_{CR} (\ell - y)\, dy = \int_0^{\ell} q\, dy$$: FORCE ACTING ON THE STEM

(FOR $\theta = 90°$ THE PISTON ACCELERATION CAN BE NEGLECTED)

• THE CONNECTING ROD CAN BE MODELED AS A BEAM, WITH CONSTANT CROSS SECTION, LOADED BY THE FOLLOWING:



$$q = \frac{\omega^2 \Lambda}{\sqrt{1-\Lambda^2}} \rho A_{CR} (\ell - y)$$: TRIANGULAR LOAD DISTRIBUTION

$$q_{MAX} = \frac{\omega^2 \Lambda}{\sqrt{1-\Lambda^2}} \rho A_{CR} \ell$$ (IN $y=0$ ⟺ BIG EYE CENTER)

• THE MAXIMUM BENDING MOMENT DUE TO THE WHIPLASH CAN BE COMPUTED AS:

$$M_{WPL} = q_{MAX} \frac{\ell^2}{9\sqrt{3}} = \frac{\omega^2 \Lambda}{\sqrt{1-\Lambda^2}} \rho A_{CR} \frac{\ell^3}{9\sqrt{3}}$$: MAXIMUM BENDING MOMENT DUE TO WHIPLASH

• AND THE CORRESPONDING STRESS:

$$\sigma_{WPL} = \frac{M_{WPL}}{W_{CR}} = \frac{\omega^2 \Lambda}{\sqrt{1-\Lambda^2}} \rho \frac{A_{CR}}{W_{CR}} \frac{\ell^3}{9\sqrt{3}}$$

W_{RC} : BENDING RESISTANCE MODULUS OF THE MOST STRESSED SECTION OF THE STEM WHICH IS AT A DISTANCE:

$$y_{MWPL,MAX} = \ell - \frac{\ell}{\sqrt{3}}$$ (FROM THE CENTER OF THE BIG EYE ⟺ $y=0$)($y = 0.42\ell$)

$\left(\xi_{MWPL,MAX} = \frac{\ell}{\sqrt{3}}\right)$

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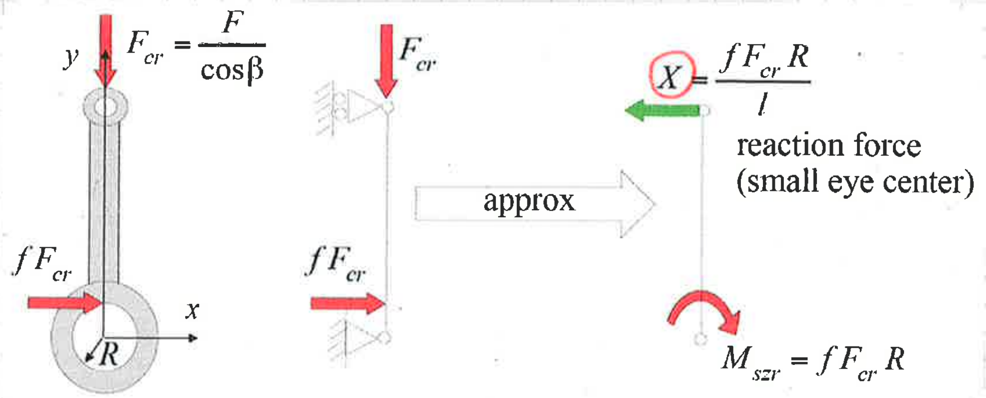
- THE BENDING OF THE STEM DUE TO SEIZURE OF THE BUSHING IN THE BIG EYE CAN BE EVALUATED BY MODELING THE CONNECTING ROD AS A BEAM LOADED BY FORCE F_{CR} ALONG THE STEM AXIS AND FRICTION FORCE ($f F_{CR}$) ON THE INTERNAL SURFACE OF THE BIG EYE.

R : BUSHING RADIUS

f : FRICTION COEFF.



- THE BENDING MOMENT ALONG THE STEM IS APPROXIMATED TO A TRIANGULAR DISTRIBUTION WITH ITS MAXIMUM VALUE ($M_{SZR} = f F_{CR} R$) IN THE BIG EYE CENTER.
- IN THE MOST STRESSED SECTION OF THE STEM IDENTIFIED FOR THE WHIPLASH EFFECT, WHICH IS AT A DISTANCE $Y_{MWPL} = \ell - \frac{\ell}{\sqrt{3}}$ FROM THE CENTER OF THE BIG EYE, THE STRESS DUE TO THE SEIZURE OF THE BUSHING CAN BE COMPUTED AS:

$$\sigma_{SZR} = \frac{M_{SZR}}{W_{CR}} \frac{\ell - Y_{MMAX}}{\ell} = \frac{M_{SZR}}{W_{CR}} \frac{\ell - (\ell - \ell/\sqrt{3})}{\ell} = \frac{M_{SZR}/\sqrt{3}}{W_{CR}} = \frac{f F_{CR} R}{W_{CR}\sqrt{3}}$$

- THE TOTAL EQUIVALENT STRESS ACTING IN THE MOST STRESSED SECTION OF THE STEM IS THE SUMMATION OF TENSILE, WHIPLASH AND SEIZURE STRESSES AND FROM A STATIC POINT OF VIEW IT HAS TO BE:

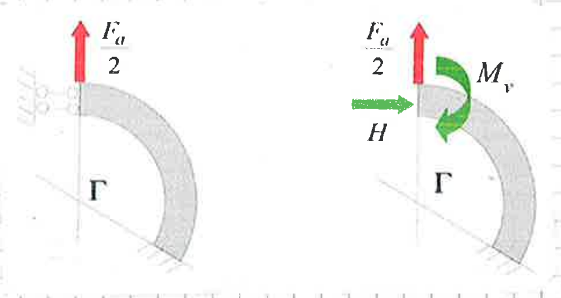
$$\sigma_{EQ,STEM} = \sigma_t + \sigma_{WPL} + \sigma_{SZR} \leq \sigma_{ADM} \quad , \sigma_{ADM} = f(\text{MATERIAL})$$

- FROM A FATIGUE POINT OF VIEW, THIS EQUIVALENT STRESS IS CYCLIC IN TIME THEN A FATIGUE VERIFICATION HAS TO BE MADE USING THE HAIGH DIAGRAM.

$\sigma_m = 0 \quad , \quad \sigma_a = \sigma_{EQ,STEM}$

$$\left(\begin{array}{l} M_{SZR} = f F_{CR} R + X \ell \quad \Rightarrow X = \frac{f F_{CR} R}{\ell} \\ M'_{SZR} : \ell - Y_{MMAX} = M_{SZR} : \ell \quad \Rightarrow M'_{SZR} = M_{SZR} \frac{\ell - Y_{MMAX}}{\ell} \Rightarrow \sigma_{SZR} = \frac{M'_{SZR}}{W_{CR}} \end{array} \right)$$

2) TO TAKE INTO ACCOUNT THE EFFECT OF THE CURVATURE, THE SMALL EYE CAN BE MODELED AS A CURVED BEAM (HALF MODEL), WHOSE OPENING ANGLE (Γ) DEPENDS ON THE GEOMETRY OF THE SMALL EYE, LOADED BY THE ALTERNATE INERTIAL FORCE.



- FROM THE CURVED BEAM THEORY THE REACTION FORCE (H) AND MOMENT (M_v) ARE:

$$H = F_a \cdot \frac{2\sin\Gamma + \Gamma\sin^2\Gamma - \sin 2\Gamma}{2\Gamma^2 + \Gamma\sin 2\Gamma - 4 + 4\cos^2\Gamma} \propto F_a \cdot f(\Gamma) \qquad H \propto F_a \cdot f(\Gamma)$$

$$M_v = \frac{F_a D_{se}}{2} \cdot \frac{\sin\Gamma - \Gamma + \Gamma\cos\Gamma - \frac{\sin 2\Gamma}{2}}{2\Gamma^2 + \Gamma\sin 2\Gamma - 4 + 4\cos^2\Gamma} \propto \frac{F_a}{2} D_{se}\, g(\Gamma)$$

- THE RESULTING BENDING MOMENT ($M(\gamma)$) IS A FUNCTION OF THE ANGULAR COORDINATE:

$$M(\gamma) = M_v + H \frac{D_{se}}{2}\cos\gamma + \frac{F_a}{4} D_{se}\sin\gamma$$

- BY INDICATING AS (M_{MAX}) THE MAXIMUM BENDING MOMENT, THAT OCCURS AT A SPECIFIC OPENING ANGLE (γ_{MAX}) OF THE SMALL EYE, THE CORRESPONDING STRESS DUE TO THE SMALL EYE CURVATURE EFFECT IS:

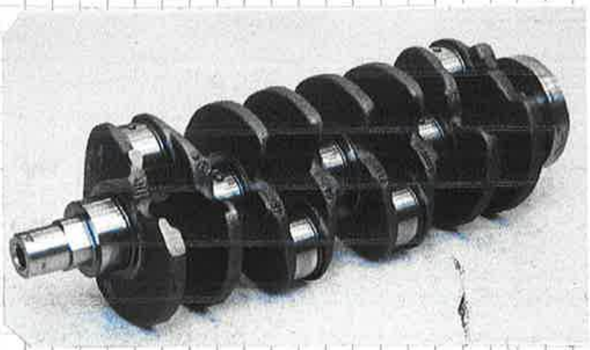
$$\sigma_{curv} = \frac{M_{MAX}}{W_{SE}}$$

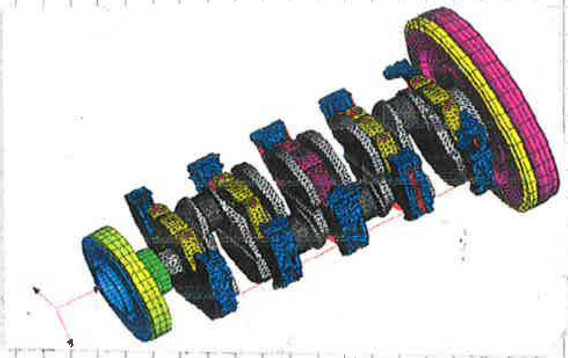
W_{SE}: BENDING RESISTANCE MODULUS OF THE SMALL EYE

- THE TOTAL EQUIVALENT STRESS ACTING IN THE MOST STRESSED SECTION OF THE SMALL EYE IS THE SUMMATION OF CIRCUMFERENTIAL, BUSHING FORCE FITTING AND CURVATURE STRESSES:

$$\sigma_{EQ,SE} = \sigma_{CIRC} + \sigma_{FIT} + \sigma_{CURV} \leq \sigma_{ADM} \quad , \quad \sigma_{ADM} = f(\text{MATERIAL}) = \frac{R_{p02}}{S_F}$$

E6 CRANKSHAFT





- INTRODUCTION
- ARCHITECTURE AND GEOMETRY
- DESIGN GUIDELINES
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  - DG - MAIN JOURNAL AND CRANKPIN LENGTHS
  - DG - $F_{t,max}$ CONFIGURATION
- STRUCTURAL VERIFICATION
  - SV - TDC CONFIGURATION
- STATIC ANALYSIS
- DYNAMIC ANALYSIS
- APPROXIMATIONS OF STATIC AND DYNAMIC ANALYTICAL APPROACH
- ANALYTICAL DYNAMIC STUDY
  - ADS - 1. LUMPED PARAMETER MODEL DEFINITION
  - ADS - 2. FREE RESPONSE AND FORCED RESPONSE
  - ADS - 3. HARMONIC ANALYSIS OF ENGINE TORQUE
  - ADS - 4. ENGINE TORQUE RESONANT HARMONICS
  - ADS - 5. MULTI-CYLINDER ENGINE
  - ADS - 5.a STAR OF THE CRANKS
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  - ADS - 5. MULTI-CYLINDER ENGINE - CONSIDERATIONS
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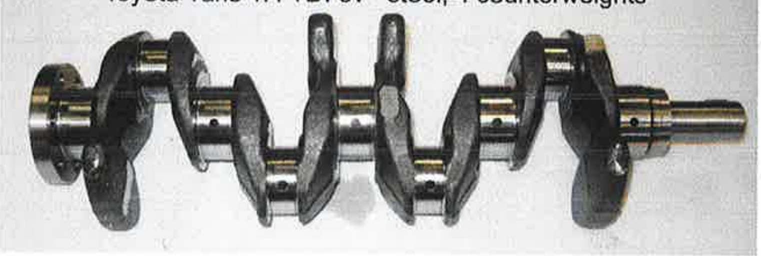
- DYNAMIC NUMERICAL ANALYSIS (FEA - MBA)
  - DNA - EXCITATION SOURCES
  - DNA - SENSOR POINTS FOR THE MB MODEL RESULTS
  - DNA - CAMPBELL DIAGRAM (CRANKSHAFT TORSION)
  - DNA - AXIAL DISPLACEMENT WITH RESPECT TO THE FLYWHEEL
  - DNA - CAMPBELL DIAGRAM (FLYWHEEL INFLECTION)
  - DNA - WATERFALL DIAGRAM

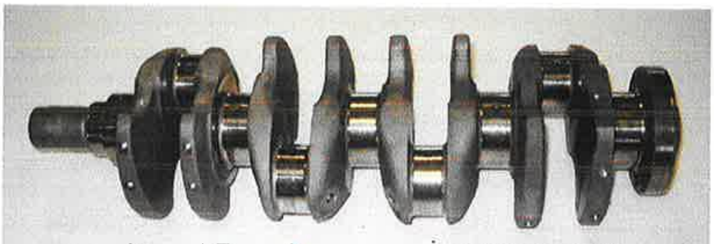
- CRANKSHAFT IS COMMONLY MADE BY DUCTILE CAST IRON (LOW/MEDIUM STRESSED ENGINES) OR FORGED/NITRIDING STEEL (C50, 40CrMo4, 38MnVS6) (HIGH STRESSED ENGINES). (CARBON OR ALLOYED STEELS)

- TO DECREASE THE ROTATING MASSES, CRANKSHAFT CAN BE LONGITUDINALLY DRILLED (HALLOW CRANKSHAFT) (EXPENSIVE SOLUTION).
- A FIRST CLASSIFICATION IS BASED ON THE COUNTERWEIGHTS NUMBER:
  - 1 COUNTERWEIGHT FOR EACH CRANK IS A LOW COST SOLUTION, BUT STIFFER CYLINDER BLOCK IS NEEDED.
  - 2 COUNTERWEIGHTS FOR EACH CRANK IS A SOLUTION IN WHICH THE CRANKSHAFT REMAINS MORE STRAIGHT, LOADS LESS THE MAIN BEARINGS AND NVH PROBLEMS DECREASE (EXPENSIVE SOLUTION).

NVH: NOISE-VIBRATION-HARSHNESS

Toyota Yaris 1.4 TDI 8v - steel, 4 counterweights





Isuzu 1.7 CDTI 16v - steel, 8 counterweights

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- ON THE CRANKSHAFT ARE ALSO MOUNTED DEVICES FOR PROVIDING TO THE ECU (ENGINE CONTROL UNIT) SIGNALS PROPORTIONAL TO THE CRANKSHAFT INSTANTANEOUS POSITION (IN TERMS OF ENGINE VELOCITY AND ACCELERATION)

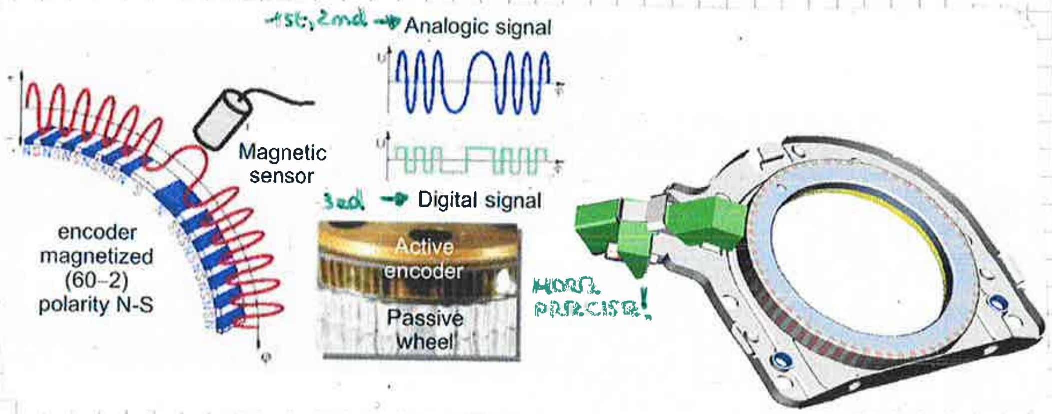
- 1st A FIRST SOLUTION USES A GEARED WHEEL WITH A READING MAGNETIC SENSOR AND WITH THE WHEEL MOUNTED ON THE CRANKSHAFT OR ON THE ENGINE FLYWHEEL.
- THE COMMON CONFIGURATION IS MADE BY A METALLIC DISC WITH (60-2) OUTER TEETH OBTAINED BY MECHANICAL CUTTING OR MACHINING.
- THE TWO "MISSING" TEETH ARE USED TO GIVE INFORMATION ABOUT THE TDC POSITION, BECAUSE TEETH ARE READ AS MAGNETIC FIELD VARIATION BY THE SENSOR RADIALLY FACED TO THEM.

Alfa Romeo V6

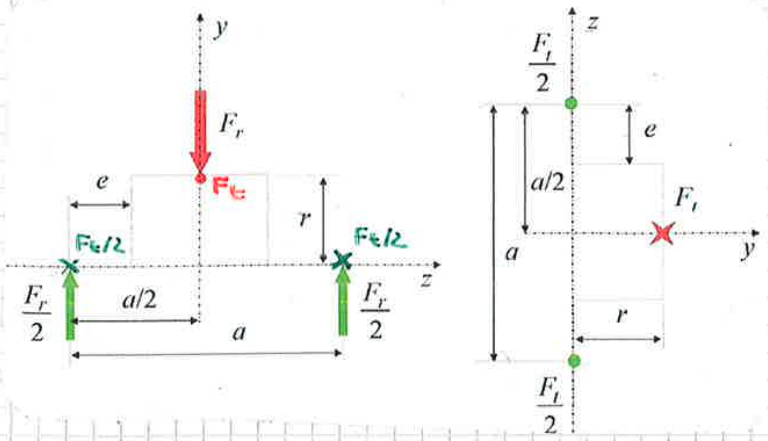
- 2nd ANOTHER SOLUTION USES (60-2) HOLES RADIALLY DRILLED ON THE PERIPHERY OF THE METALLIC DISC.
- 3rd A THIRD AND MORE PRECISE SOLUTION USES A DIGITAL DEVICE MADE BY A MAGNETIC METALLIC RING EXTERNALLY COATED WITH MAGNETIZED RUBBER AND WITH (60-2) POLARITY N-S; THE MAGNETIC SENSOR FACED TO THE RING READS THE POLARITY PASSAGE AND ACQUIRES THE CRANKSHAFT INSTANTANEOUS POSITION.

FIAT SDE 1.3

Q.: WHY "ONLY" 60-2 TEETH/HOLES/POLARITY PASSAGES ?
360-2 ISN'T POSSIBLE, IS IT ?



- COMPUTATIONAL MODEL SCHEME:



WITH:
× LOAD (× : "ENTERING")
● CONSTRAINTS (HINGE, SUPPORT)
(⊥ TO THE DRAWING PLANE)
YZ: LATERAL PLANE
ZY: PERPENDICULAR PLANE

- THE CRANKPIN JOURNAL IS LOADED BY TWO BENDING MOMENTS (M_y, M_x) (M_y DUE TO TANGENTIAL FORCE, M_x DUE TO RADIAL FORCE) AND ONE TORQUE (M_z) (M_z DUE TO TANGENTIAL FORCE):

$$M_y = \frac{F_t}{2}\frac{a}{2} \; ; \; M_x = \frac{F_R}{2}\cdot\frac{a}{2} \; ; \; M_z = \frac{F_t}{2}\cdot R$$

WITH
$$\begin{cases} F_t = F_{tR}\sin(\theta+\beta) \\ F_R = F_{tR}\cos(\theta+\beta) \\ F_{tR} = \dfrac{F}{\cos\beta} = \dfrac{F_g+F_a}{\cos\beta} \end{cases}$$

- BENDING STRESS (σ_b):

$$\sigma_b = \frac{M_b}{W_b} = \frac{\sqrt{M_x^2+M_y^2}}{W_b} = \frac{a}{4}\frac{\sqrt{F_R^2+F_t^2}}{W_b}$$

(REMEMBER THAT $M_y = M_{by}$ <=> AROUND y)

- SHEAR STRESS (τ):

$$\tau = \frac{M_z}{W_t} = \frac{F_t R}{2 W_t}$$

- COMPARING VON MISES EQUIVALENT STRESS WITH THE MATERIAL ADMISSIBLE STRESS (σ_{ADM}) (CAST IRON 60 [MPa], STEEL 100 [MPa] : THESE VALUES IN ORDER TO CONSIDER ALSO THE FATIGUE)

$$\sigma_{EQ} = \sqrt{\sigma_b^2+3\tau^2} \leq \sigma_{ADM} \implies a$$

- THE MAIN JOURNAL IS LOADED BY TWO BENDING MOMENTS (M_y, M_x) (M_y DUE TO TANGENTIAL FORCE, M_x DUE TO RADIAL FORCE) AND ONE TORQUE (M_z) (M_z DUE TO TANGENTIAL FORCE):

$$M_y = \frac{F_t}{2}\cdot e \; ; \; M_x = \frac{F_R}{2}\cdot e \; ; \; M_z = \frac{F_t}{2} R$$

- BENDING STRESS (σ_b):

$$\sigma_b = \frac{M_b}{W_b} = \frac{\sqrt{M_x^2+M_y^2}}{W_b} = \frac{e}{2}\frac{\sqrt{F_R^2+F_t^2}}{W_b}$$

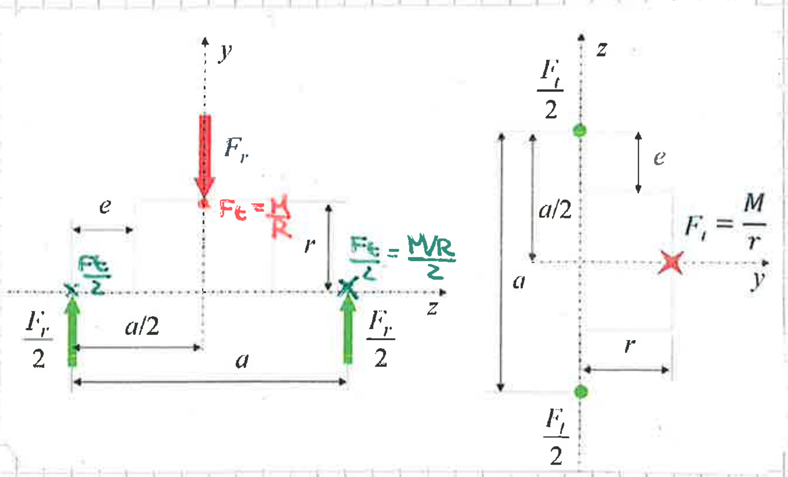
- SHEAR STRESS (τ):

$$\tau = \frac{M_z}{W_t} = \frac{F_t R}{2 W_t}$$

- COMPARING VON MISES EQUIVALENT STRESS WITH THE MATERIAL ADMISSIBLE STRESS (σ_{ADM}) (CAST IRON 60 [MPa], STEEL 100 [MPa]):

$$\sigma_{EQ} = \sqrt{\sigma_b^2+3\tau^2} \leq \sigma_{ADM} \implies e$$

- THE COMPUTATIONAL MODEL IS THE SAME OF $F_{t,MAX}$ CONFIGURATION, WITH DIFFERENT FORCES AND CONSTRAINT REACTIONS.



WITH:
× LOAD
● CONSTRAINTS
(⊥ TO THE DRAWING PLANE)

- SV - TDC CONFIGURATION

- CRANKPIN JOURNAL: IT IS LOADED BY ONE BENDING MOMENT (M_x DUE TO RADIAL FORCE) AND ONE TORQUE (M_z DUE TO THE "EQUIVALENT TANGENTIAL FORCE")

$M_x = \frac{F_R}{2}\frac{a}{2}$; $M_z = \frac{F_t}{2} R$ (NO M_y)

- MAIN JOURNAL: IT IS LOADED BY ONE BENDING MOMENT (M_x DUE TO RADIAL FORCE) AND ONE TORQUE (M_z DUE TO THE "EQUIVALENT TANGENTIAL FORCE").

$M_x = \frac{F_R}{2} \cdot e$; $M_z = \frac{F_t}{2} \cdot R$ (NO M_y)

- CHEEK: THE SAME LOAD CONDITION OF $F_{t,MAX}$ CONFIGURATION ACTS ON THE CHEEK:

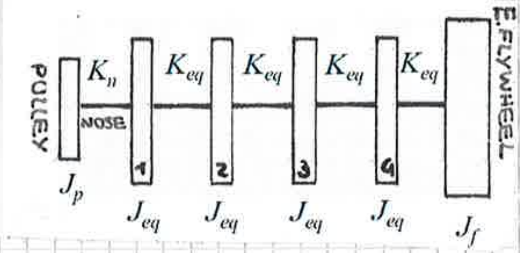
$N = \frac{F_R}{2}$; $M_x = \frac{F_R}{2} \cdot e$; $M_z = \frac{F_t}{2} \cdot R$

- THE VERIFICATION OF EACH PART FOLLOWS THE GUIDELINES USED IN THE $F_{t,MAX}$ CONFIGURATION TO SIZE THE CRANK DIMENSIONS (NOW KNOWN VALUES).

- WE HAVE ANALYZED A GENERIC CRANK; NOW WE HAVE TO VERIFY THE CRANKSHAFT AS A COMPLETE COMPONENT.
- THE CRANKSHAFT IS AN HYPERSTATIC COMPONENT BECAUSE IT HAS 3 D.O.F AND THE CONSTRAINTS ARE 6 ⇒ 3 DEGREES OF HYPERSTATIC BEHAVIOUR.
- THE STATIC ANALYTICAL SOLUTION (STATIC ANALYSES) FIXES THE ATTENTION AND SOLVES THE COMPUTATION OF THE CRANKSHAFT ONLY IN TERMS OF BENDING.
- IF WE WANT TO TAKE INTO ACCOUNT ADDITIONAL EFFECTS, SHEAR AND TORSIONAL, WE HAVE TO PASS FROM AN ANALYTICAL METHODOLOGY TO A NUMERICAL ONE (FE INVESTIGATION) THROUGH A FE CODE.

• DYNAMIC ANALYSIS

• TO COMPUTE THE TORSIONAL STRESSES ACTING ON THE CRANKSHAFT, A LUMPED PARAMETER SYSTEM TORSIONALLY EQUIVALENT (FROM A DYNAMIC POINT OF VIEW) TO THE CRANKSHAFT IS CONSIDERED.



• EACH CRANK IS MODELED AS AN INFINITELY STIFF FLYWHEEL (WITH AN EQUIVALENT DYNAMIC MOMENT OF INERTIA J_{EQ}) PLUS A MASSLESS TORSIONAL BAR (WITH TORSIONAL STIFFNESS K_{EQ} EQUIVALENT TO THE CRANK STIFFNESS).

• TWO ADDITIONAL FLYWHEELS CAN APPEAR IN THE MODEL: ONE REPRESENTS THE ENGINE FLYWHEEL (J_f), THE OTHER REPRESENTS THE PULLEY (J_P); THE BAR K_m SIMULATES THE CRANKSHAFT NOSE.

• THE METHODOLOGY TO INVESTIGATE THE TORSIONAL DYNAMIC BEHAVIOUR OF THE CRANKSHAFT CONSISTS OF THE FOLLOWING STEPS:

1) DEFINITION OF THE EQUIVALENT LUMPED PARAMETER SYSTEM;
2) DYNAMIC ANALYSIS OF THE FREE BEHAVIOUR OF A SINGLE-CYLINDER ENGINE (TORSIONAL NATURAL FREQUENCIES AND CORRESPONDING MODE SHAPES);
3) FOURIER DECOMPOSITION OF GAS AND RECIPROCATING INERTIAL FORCES;
4) IDENTIFICATION OF THE ENGINE TORQUE HARMONICS THAT ARE IN RESONANT CONDITION (CAMPBELL DIAGRAM); (ENG. TORQUE HARM. COMPARED WITH NATURAL FREQ.)
5) ENGINE TORQUE ON THE j-th CYLINDER OF THE MULTI-CYLINDER ENGINE:
   a. PHASE DIAGRAM OF THE CRANKS (OR STAR OF THE CRANKS).
   b.* PHASE DIAGRAMS OF THE HARMONICS (OR STARS OF THE HARMONICS).
6) COMPUTATION OF THE FORCED RESPONSE UNDER RESONANT CONDITION AND CORRESPONDING TORSIONAL DYNAMIC STRESSES, THEN COMPARED WITH THE STRENGTH OF THE MATERIAL ($R_{p0.2}$).

b.* TO TAKE INTO ACCOUNT THE EFFECTS OF ALL CYLINDERS.

- TO STUDY THE DYNAMIC BEHAVIOUR OF THE EQUIVALENT SYSTEM, THE EQUATION OF MOTION HAS TO BE SOLVED (FREE BEHAVIOUR + FORCED RESPONSE):

$$[J]\{\ddot{\phi}\} + [K]\{\phi\} = \{M(\Theta)\}$$

WITH:

$[J]$: DIAGONAL INERTIA MATRIX $\nu \times \nu$ ($\nu = z+2$ NUMBER OF TORSIONAL DOF)

$[K]$: TRIDIAGONAL SYMMETRIC STIFFNESS MATRIX $\nu \times \nu$

$\{\phi\}$: TORSIONAL DOF / AMPLITUDE (OF FORCED R. FOR EX.)

$\{M(\Theta)\}$: ENGINE TORQUE

- THE INERTIA MATRIX $[J]$ HAS CONSTANT COEFFICIENTS OVER TIME ONLY IF CONSTANT OVER TIME EQUIVALENT MOMENT OF INERTIA AND STIFFNESS ARE COMPUTED FOR EACH CRANK.
- DAMPING WILL BE TAKEN INTO ACCOUNT IN TERM OF AN EQUIVALENT VISCOUS DAMPING THAT APPEARS DIRECTLY IN THE EXPRESSION OF THE FORCED RESPONSE.
- THE EQUIVALENT MOMENT OF INERTIA (J_{EQ}) OF THE GENERIC CRANK IS THE SUMMATION OF THE INERTIAS OF CRANK, ROTATING AND RECIPROCATING MASSES OF CONNECTING ROD, AND AUXILIARY INERTIA OF THE CONNECTING ROD REDUCED MODEL (FROM THE CRANK MECHANISM ANALYSIS).

- THE AVERAGE EQUIVALENT MOMENT OF INERTIA OF THE SINGLE CRANK ($\bar{J}_{EQ}$) IS THEN: (THEORICALLY $J_{EQ} \neq$ CONST, BUT WE ASSUME A CONSTANT AVERAGE $\bar{J}_{EQ}$)

$$\bar{J}_{EQ} = J_C + m_{CR,R} R^2 + (m_{CR,a} + m_P)\frac{8+2\Lambda^2+\Lambda^4}{16}R^2 + J_0 \frac{4\Lambda^2+\Lambda^4}{8}$$ (MEAN VALUE OF THE J_{EQ})

EVENTUALLY WE CAN ADD ALSO m_{WP} (BETTER)

WITH:

$\bar{J}_{EQ}$: EQUIVALENT MOMENT OF INERTIA OF THE SINGLE CRANK

J_C : CRANK MOMENT OF INERTIA

$m_{CR,R}$: CONNECTING ROD MASS CONCENTRATED IN THE BIG EYE

$m_{CR,a}$: CONNECTING ROD MASS CONCENTRATED IN THE SMALL EYE

m_P : PISTON MASS

J_0 : AUXILIARY MOMENT OF INERTIA OF THE CONNECTING ROD.

- THE EQUIVALENT MOMENT OF INERTIA (J_{EQ}) IS CONSIDERED AVERAGE ON THE ENGINE CYCLE AND THEN CONSTANT IN TIME.
- ASSUMING $\Lambda = 0$, A SIMPLIFIED FORM OF THE AVERAGE EQUIVALENT MOMENT OF INERTIA CAN BE OBTAINED:

$$\bar{J}_{EQ} \approx J_C + m_{CR,R} R^2 + \frac{m_{CR,a} + m_P}{2} R^2$$

- ADS - 2. FREE RESPONSE

- TO STUDY THE FREE RESPONSE OF THE EQUIVALENT SYSTEM, THE UNLOADED EQUATION OF MOTION IS CONSIDERED:

$$[J]\{\ddot{\phi}\} + [K]\{\phi\} = \{0\}$$

- TORSIONAL NATURAL FREQUENCIES (EIGENVALUES) AND CORRESPONDING VIBRATION MODES (EIGENVECTORS) CAN BE COMPUTED SOLVING THE EIGENPROBLEM:

$$\{\phi\} = \{\phi_0\} e^{i\lambda_m t} \; ; \; \{\dot{\phi}\} = i\lambda_m \{\phi_0\} e^{i\lambda_m t} \; ; \; \{\ddot{\phi}\} = -\lambda_m^2 \{\phi_0\} e^{i\lambda_m t}$$

classical Eigenv. probl. $\rightarrow \lambda_m$

$$-[J]\lambda_m^2\{\phi_0\}e^{i\lambda_m t} + [K]\{\phi_0\}e^{i\lambda_m t} = \{0\} \iff [-[J]\lambda_m^2 + [K]] \cdot \{\phi_0\} e^{i\lambda_m t} = \{0\} \quad \det[-[J]\lambda_m^2 + [K]] = 0$$

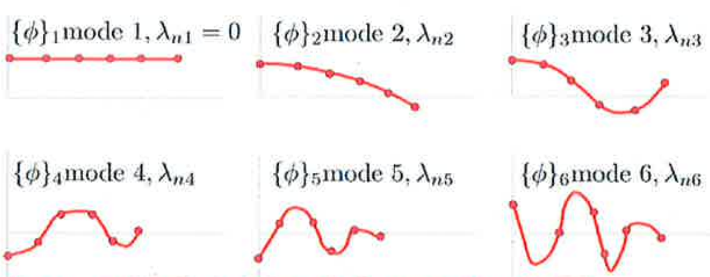
λ_{mi} : i-th TORSIONAL NATURAL FREQUENCY ($i = 1, V$ WITH $\lambda_{m1} = 0$ RIGID MODE)

$\{\phi\}_i$: i-th TORSIONAL MODE

- AS THE EQUIVALENT MOMENTS OF INERTIA OF THE CRANKS ARE ASSUMED CONSTANT OVER TIME, THE NATURAL FREQUENCIES ARE CONSTANT AND INDEPENDENT ON CRANK ANGLE θ, AND THEN ON CRANKSHAFT SPIN SPEED ω.
- CONSEQUENTLY, ON THE CAMPBELL DIAGRAM (λ VS ω) THE NATURAL FREQUENCIES WILL BE REPRESENTED AS HORIZONTAL STRAIGHT LINES.

$V = Z = 6 \Rightarrow 6$ MODES

Torsional modes



6 CYL. → 6 DOF → 6 λ_m → 6 VIBRATION MODES

PREVIEW: DAMPING IS IMPORTANT WHEN WE INVESTIGATE THE FORCED RESPONSE. IN THIS CASE WE ADD THE DAMPING EFF. NOT DIRECTLY IN THE EQUATION OF MOTION, BUT EXPRESSING THE AMPLITUDE OF THE FORCED RESPONSE WITH AN EXPRESSION WHICH CONSIDER DAMPING.

- ADS - 2. FORCED RESPONCE

- TO STUDY THE FORCED RESPONCE OF THE EQUIVALENT SYSTEM, THE HARMONIC ANALYSIS OF THE ENGINE TORQUE (GAS + INERTIA) HAS TO BE PERFORMED.
- FOR THE GENERIC SINGLE CYLINDER THE ENGINE TORQUE CAUSED BY GAS AND RECIPROCATING INERTIAL FORCES CAN BE DEVELOPED IN FOURIER SERIES AS A SUM OF HARMONICS:

$$M(\theta) = M_g(\theta) + M_a(\theta)$$

WITH: GENERIC FUNCTION; STATIC COMP. OF THE GASES: IT DOESN'T PRODUCE TORSIONAL VIBRATION; IT'S RELATED TO THE E. POWER; MEAN VALUES $\neq 0$; HARMONIC NUMBER (K)

$$M_g(\theta) = P_g(\theta)\frac{\pi D^2}{4} \cdot f_1(\theta) \cdot R \cong A_0 + \sum_{k=1}^{20} A_k \cos(k\omega' t) + \sum_{k=1}^{20} B_k \sin(k\omega' t)$$

$K = 1, \dots, 20$ SUFF. FOR GAS F.

$$M_a(\theta) = -m_a\omega^2 R \cdot f_2(\theta) \cdot R \cong \omega^2 \sum_{k=1}^{8} A_k^* \cos(k\omega t) + \omega^2 \sum_{k=1}^{8} B_k^* \sin(k\omega t)$$

MEAN VALUE = 0

$K = 1, \dots, 8$ SUFF. FOR R.I.F.

ω': PULSATION OF THE FUNDAMENTAL HARMONIC ; ω: SPIN SPEED OF ENGINE

$$\begin{cases} \omega' = \omega/2 & \text{FOR 4 STROKE ENGINE (4T)} \\ \omega' = \omega & \text{FOR 2 STROKE ENGINE (2T)} \end{cases}$$

A_0, A_k, B_k (A_k^*, B_k^*) : KNOWN TERMS REPRESENTING AMPL. INTEGRAL REPRES. OF FOURIER COMP. DEPENDING ON ANGLE θ.

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- THE FOURIER SERIES OF THE **TOTAL ENGINE TORQUE** ACTING **ON THE SINGLE CYLINDER** CAN BE OBTAINED COMBINING THE VECTORS THAT REPRESENT THE HARMONICS OF THE GAS AND THE INERTIAL RECIPROCATING ACTION:

$$M(\theta) = \sum_{k=1}^{20} \left(M_{gk} e^{i(k\omega' t + \phi_{gk})} + \omega^2 M_{ak} e^{i(k\omega' t + \phi_{ak})} \right)$$

OVERALL CONSIDERING ONLY THE IMPORTANT FOR THE TORSIONAL DYN. BEHAVIOUR WITHOUT CONSTANT TERM A_0.

4T: $f_g = \frac{f_a}{2}$ $\left(f = \frac{\omega}{2\pi} \right)$

WITH:

M_{gk}, ϕ_{gk}: AMPLITUDE AND PHASE OF THE k-th HARMONIC OF THE **GAS TERM**

M_{ak}, ϕ_{ak}: AMPLITUDE AND PHASE OF THE k-th HARMONIC OF THE **INERTIA TERM**

(IN CASE OF **CENTERED CRANK MECHANISM**, ALL $\phi_{ak} = \pi/2$)

- ADS - 4. ENGINE TORQUE RESONANT HARMONICS

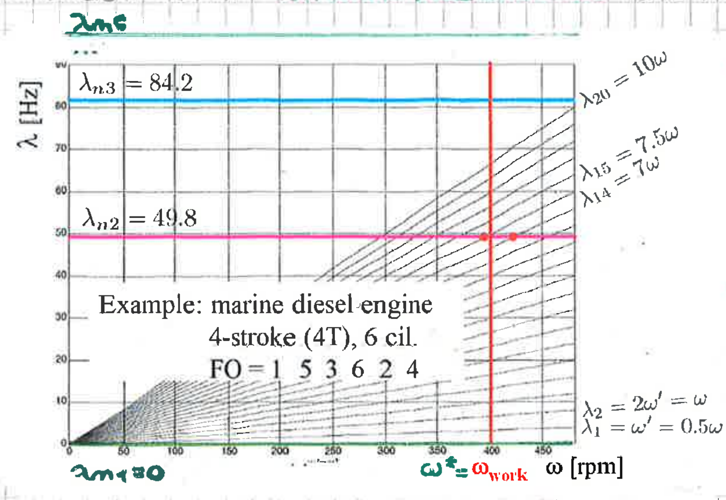
- WHEN THE **FREQUENCY OF ONE** OF THE ENGINE **TORQUE HARMONICS** COINCIDES WITH **ONE OF** THE **CRANKSHAFT NATURAL FREQUENCIES**, THERE IS **RESONANCE** BETWEEN **EXCITATION** AND **ONE OF THE NATURAL MODES** OF THE CRANKSHAFT.
- **TO IDENTIFY** THE **HARMONICS** OF THE ENGINE TORQUE **THAT ARE RESONANT** WITH THE CRANKSHAFT NATURAL FREQUENCIES, THE **CAMPBELL DIAGRAM** (λ VS ω) IS **USED**.
- THE TORSIONAL **NATURAL FREQUENCIES** λ_{mi} $(i = 1, V)$ APPEAR **ON THE CAMPBELL DIAGRAM** AS **HORIZONTAL STRAIGHT LINES** (AS THEY ARE **INDEPENDENT FROM** THE **SPIN SPEED**).
- THE PULSATION OF THE GENERIC k-th HARMONIC OF THE ENGINE TORQUE DEPENDS ON TIME:

$$\lambda_k = k \cdot \omega' \quad ; \; k = 1, \dots, 20 \quad ; \quad \omega' = \begin{cases} \omega/2 & \text{FOR 4T ENGINE} \\ \omega & \text{FOR 2T ENGINE} \end{cases}$$

λ_k: **ENGINE TORQUE HARMONIC FREQ.**

THEREFORE, THE ENGINE TORQUE **HARMONIC FREQUENCIES** λ_k APPEAR AS **STRAIGHT LINES STARTING FROM THE ORIGIN** OF THE **CAMPBELL DIAGRAM.**

- FOR A SPECIFIED ENGINE "WORKING" SPEED ω_{work}, THE **RESONANT HARMONICS** OF THE ENGINE TORQUE **ARE THOSE** THAT ARE **NEAR THE INTERSECTION** BETWEEN THE CRANKSHAFT **NATURAL FREQUENCIES** AND THE **VERTICAL LINE** OF ω_{work}



→ FOR A SPECIFIC SPEEN SPEED (ω^*) WE HAVE A POSSIBLE RESONANCE OF THE 2nd MODE WITH RESPECT TO TWO HARMONICS OF THE E. TORQUE

→ ALL THESE REPRESENTATIONS ARE RELATED TO AN **IDEAL MONO-CYL. SIT.** ONLY THE COMP. OF λ_m COMES FROM THE ACTUAL PRESSURE OF 1 CYL. BUT THE COMP. OF M (SO THE EXC.) IS RELATED TO ONE

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- ADS - 5.6. STARS OF THE HARMONICS

- THE **PHASE SHIFT** ($\delta_{k,j}$) IS GIVEN BY THE **PRODUCT** BETWEEN THE **HARMONIC ORDER** (ORD)

$$ORD = \begin{cases} k/2 & \text{FOR 4T ENGINE} \\ k & \text{FOR 2T ENGINE} \end{cases}$$

AND THE **ANGULAR PHASE SHIFT** (Ψ) (ANGLE BETWEEN CRANKS)

$$\delta_{k,j} = ORD \cdot \Psi = \begin{cases} \frac{k}{2}\Psi & \text{FOR 4T ENGINE} \\ k\Psi & \text{FOR 2T ENGINE} \end{cases}$$

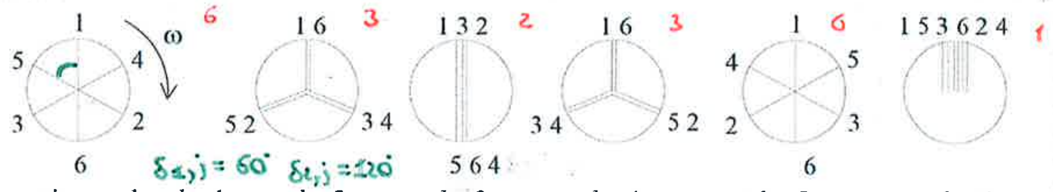
j-th CYLINDERS UNDER INVESTIGATION, ON WHICH THE k-th HARMONIC OF THE ENGINE TORQUE ACTS.

- BY KNOWING THE **FO**, THE **PHASE DIAGRAMS** NAMED "**STARS OF THE HARMONICS**" CAN BE DRAWN, STARTING FROM A CERTAIN POSITION OF THE FIRST VECTOR AND ROTATING IT IN **OPPOSITE DIRECTION** WITH RESPECT TO ω.

- THE **PHASE SHIFT** IS (4T): (F.O. = 1, 5, 3, 6, 2, 4)

$\delta_{k,j} = \frac{k}{2} \cdot \Psi = \frac{k}{2} \cdot 120° = k \cdot 60°$, $k = 1, \dots, 20$; $j = 1, \dots, 6$ $\qquad \Psi = \frac{4\pi}{z} = \frac{4\pi}{6} = \frac{2}{3}\pi = 120°$

THE **STARS OF THE HARMONICS** ARE THEN:



harmonic number $k=1$	$k=2$	$k=3$	$k=4$	$k=5$	$k=6$
harmonic order $\boxed{ord=0.5}$	$\boxed{ord=1}$	$ord=1.5$	$\boxed{ord=2}$	$\boxed{ord=2.5}$	$ord=3$
and also	and also	and also	and also	and also	and also
harmonic number $k=7$	$k=8$	$k=9$	$k=10$	$k=11$	$k=12$
harmonic order $\boxed{ord=3.5}$	$ord=4$	$ord=4.5$	$ord=5$	$ord=5.5$	$ord=6$
		$\delta_{3,j}=180°$	$\delta_{4,j}=60°$	$\delta_{5,j}=300°$	$\delta_{6,j}=360°$

- IT CAN BE NOTICED THAT:
  - THE **STAR OF HARMONICS** WITH **ORDER 3.5** IS **EQUAL TO** THE STAR OF THE HARMONICS WITH **ORDER 0.5**.
  - THE STAR OF THE HARMONICS WITH **ORDER 2.5** IS THE **MIRRORED** IMAGE OF THE STAR OF THE HARMONICS WITH **ORDER 0.5**.
- THE **STARS OF HARMONICS EQUAL** TO EACH OTHER AND THOSE EQUAL TO THEIR MIRRORED IMAGES ARE COMMONLY CONSIDERED TO **BELONG TO** THE **SAME GROUP**
- **FOR 4-STROKE (4T) ENGINE**, THE NUMBER OF DIFFERENT GROUPS IS:

$$\text{NUMBER OF DIFFERENT GROUPS} = \begin{cases} \frac{z+2}{2} & \text{IF } z \text{ IS EVEN} \\ \frac{z+1}{2} & \text{IF } z \text{ IS ODD} \end{cases} \qquad \left(\frac{6+2}{2} = 4 \text{ DIFF. GROUPS}\right)$$

PREVIEW: WE HAVE TO CONSIDER k OF RESONANT HARMONICS FROM CAMPBELL DIAGRAM AND k OF "MAJOR ORDERS".

- FINALLY, FOR EACH DANGEROUS HARMONIC OF THE ENGINE TORQUE, THE TOTAL TORSIONAL SHEAR STRESS ACTING ON EACH TORSIONAL BAR EQUIVALENT TO EACH CRANK CAN BE COMPUTED AS:

$$\tau_s = \tau_{STAT,s} + \tau_{DYN,s}$$

WHERE $s = 1, 2, 3, \dots, m$ ARE THE CRANKSHAFT SECTIONS OF INTEREST

- IN A MORE EXPLICIT FORM IT CAN BE WRITTEN:

$$\tau_s = s\frac{A_0}{W_{t,s}} + \frac{K_s(\phi_{s+1} - \phi_s)}{W_{t,s}}$$

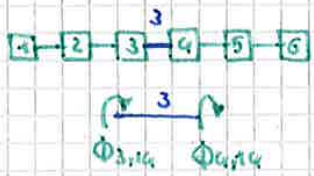
WITH:

A_0 : STATIC CONTRIBUTION OF THE GAS (S : NUMBER OF THE CRANK)

$W_{t,s}$: TORSIONAL RESISTANCE MODULUS OF THE S-th PART OF THE CRANKSHAFT

K_s : TORSIONAL STIFFNESS OF THE S-th PART OF THE CRANKSHAFT

ϕ_{s+1}, ϕ_s : AMPLITUDES OF THE FORCED RESPONSE, AT NODES (S+1) AND S, RESPECTIVELY (S+1)-th AND S-th, COMPONENTS OF VECTOR $\{\phi\}_k$.



- WE ISOLATE A BAR ($k = 14$, 6 CYL.)
- IT IS SUBJECTED TO A RELATIVE ROTATION $(\phi_{4,14} - \phi_{3,14})$

$$\tau_{DYN,3} = \frac{K_3(\phi_{4,14} - \phi_{3,14})}{W_{t,3}} \quad ; \quad \tau_{STAT,3} = \frac{3 \cdot A_0}{W_{t,3}}$$

- WE CAN COMPUTE THE TORSIONAL SHEAR STRESS (τ_s) IN ALL PARTS (CRANKS, S) of CRANKSHAFT TO IDENTIFY THE MOST STRESSED SECTION (CRANK), WITH RESPECT THE TORSIONAL BEHAVIOUR ONLY, WITH AN ANALYTICAL PROCEDURE.

1 FEA RELAXES MANY OF THE CLASSICAL APPROACH LIMITATIONS, BUT DOES NOT ALLOW TO SIMULATE THE DYNAMICS OF THE CRANKSHAFT AS A COMPONENT THAT DYNAMICALLY INTERFERS WITH THE OTHER ELEMENTS OF THE POWERTRAIN.

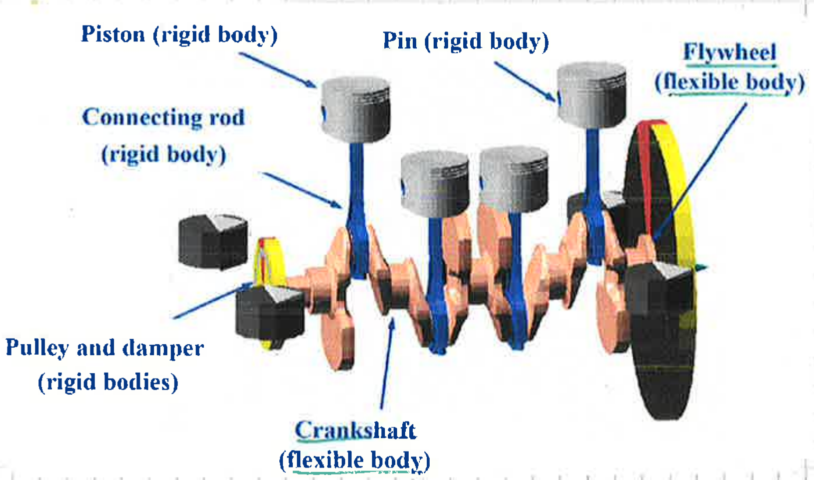
2 CRANKSHAFT GYROSCOPIC EFFECT IS NOT CONSIDERED, AND THE INERTIAL FORCES ARE CALCULATED AND APPLIED AS STATIONARY LOADS.

3 THE REYNOLDS EQUATIONS GOVERNING THE BEHAVIOUR OF THE OIL FILM OF THE BUSHINGS ARE NOT SOLVED FOR EACH TIME INSTANT, AND THE OIL FILM IS SIMULATED AS A NON-LINEAR SPRING WITH CHARACTERISTIC ASSUMED BY DEFAULT.

- TO OBTAIN A MORE DETAILED DYNAMIC INVESTIGATION OF THE CRANKSHAFT, A FE-MULTIBODY ANALYSIS (FEA-MBA) IS USED IN ORDER TO:
  1- COMPUTE INERTIAL FORCES AND AUTOMATICALLY APPLY THEM TO THE MODEL
  2- TAKE INTO ACCOUNT THE VARIABILITY OVER TIME OF THE MOMENT OF INERTIA OF THE GROUP CONNECTING ROD - CRANK MECHANISM.
  3- TAKE INTO ACCOUNT DYNAMIC INTERACTIONS AND HYDRO-DYNAMIC ELASTO-HYDRODYNAMIC LUBRICATION CONDITIONS (I.E. MAIN BEARINGS).
  4- TAKE INTO ACCOUNT THE EFFECTS OF LOGICAL GYROSCOPIC STIFFENING (E.G. NEAR THE ENGINE FLYWHEEL).

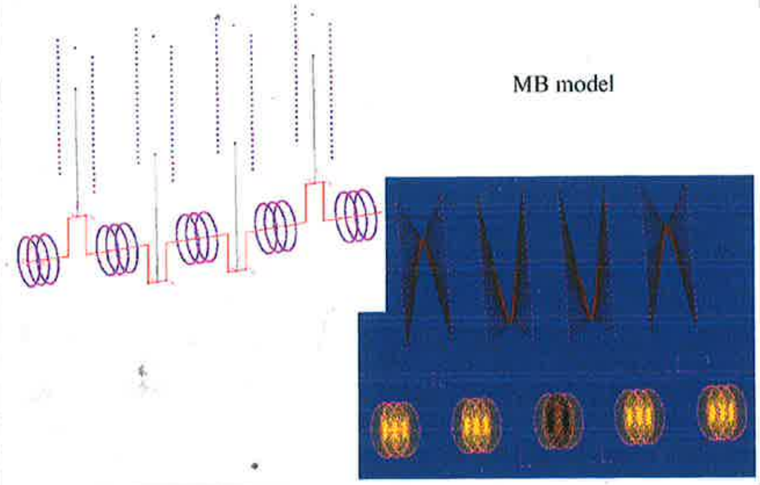
- THE FIRST STEP OF THE FEA-MBA OF A CRANKSHAFT IS THE CREATION OF THE FE MODELS OF THE VARIOUS PARTS (CRANKSHAFT, FLYWHEEL, CONNECTING RODS, CYLINDER BLOCK, ...):
  - CONNECTING RODS ARE MODELLED AS BEAM ELEMENTS WITH DYNAMIC CHARACTERISTICS EQUAL TO THE ACTUAL ONES (WITH THE SAME NATURAL FREQUENCIES).
  - PISTONS ARE ASSUMED AS CONCENTRATED MASSES AT THE CORRESPONDING SMALL EYE OF THE CONNECTING RODS.
  - THE FE MODEL OF THE CRANKSHAFT WITH ITS FLYWHEEL AND PULLEY (TOGETHER WITH THE SEISMIC MASS OF THE DYNAMIC DAMPER) IS USED TO TUNE THE DYNAMIC DAMPER AND TO IDENTIFY ITS CORRECT STIFFNESS

- TO PASS FROM THE FE MODEL TO THE MB CODE, THE FE MODEL DOF HAS TO BE STRONGLY REDUCED IN NUMBER AND THE CHOSEN MASTER DOF ARE:
  - CONSTRAINT NODES BETWEEN CRANKSHAFT AND CYLINDER BLOCK, CRANKSHAFT AND FLYWHEEL, CRANKSHAFT AND PULLEY.
  - NODES ON WHICH EXTERNAL LOADS AND REACTIONS ARE APPLIED.
  - NODES ON WHICH THE MBA RESULTS ARE COMPUTED.
- STIFFNESS AND MASS MATRICES COMPUTED BY FEA (CRANKSHAFT, FLYWHEEL, CYLINDER BLOCK, WALLS...) HAVE TO BE REDUCED BY USING DIFFERENT REDUCTION TECHNIQUES: STATIC OR GUYAN CONDENSATION, DYNAMIC OR CRAIG-BAMPTON CONDENSATIONS (COMMON IN COMMERCIAL FE CODES).
- THE REDUCED MATRICES ARE IMPORTED IN THE MB CODE AND THE KINEMATIC CHARACTERISTICS ARE APPLIED TO EACH BODY IN TERMS OF RIGID OR DEFORMABLE BODY (3 OR 6 DOF), CONDENSED OR CONSTRAINED BODY WITH IMPOSED DISPLACEMENT LAWS.



ALSO CYLINDER BLOCK IS CONSIDERED FLEXIBLE

- THE GAS PRESSURE, AVAILABLE IN TABULAR FORM, AND THE ENGINE FO ARE THEN INTRODUCED IN THE MB MODEL AND APPLIED TO THE SMALL EYE OF THE CONNECTING RODS.
- THE MB CODE CALCULATES THE FORCES ACTING ON PISTONS AND SHIFTS EACH OTHER BY THE RIGHT CRANK ANGLE.
- TO GUARANTEE THE MOMENTS EQUILIBRIUM, A TORQUE MUST BE APPLIED TO THE CRANKSHAFT (ON THE NODE REPRESENTING THE CENTRE OF THE FLYWHEEL) AND ITS VALUE IS AUTOMATICALLY COMPUTED BY THE MB CODE.
- THE PROPER SETTING OF THE MODEL PARAMETERS IN THE MB CODE (FO, RELATIVE POSITION BETWEEN CRANKS, NUMBER OF CYLINDERS, GAS PRESSURE IN TABULAR FORM, SPIN SPEED, GEOMETRIC PARAMETERS OF THE ENGINE, ...) CAN BE VERIFIED BY VIEWING THE CREATED MB MODEL.



- IN MB SIMULATION THE FIRST 5-6 ENGINE CYCLES ARE CARRIED OUT TO GET THE STEADY STATE CONDITION OF THE MODEL.
- IN FACT, DURING TRANSIENT CONDITION THE RESULTS OF THE MB MODEL ARE AFFECTED BY NUMERICAL NOISE (AS THE MODEL STARTS FROM STEADY AND HAS TO OVERCOME ALL THE INERTIAS TO MOVE TO REGIME).
- THE INTEGRATION OF THE EQUATIONS OF MOTION IS MORE DIFFICULT IN THE FIRST ITERATIONS BECAUSE THEY HAVE NOT BASELINE NUMERICAL VALUES TO START AND THE INTEGRATION ALGORITHM HAS TO STABILIZE ITSELF.
- THE SIMULATION RESULTS ARE OBTAINED IN TERMS OF DISPLACEMENTS, ANGULAR VELOCITIES AND ANGULAR ACCELERATIONS (OF THE CONDENSED NODES, I.E. MASTER DOFS), FORCES AND MOMENTS ON THE NODES REQUIRED AT THE MB MODEL SET UP.

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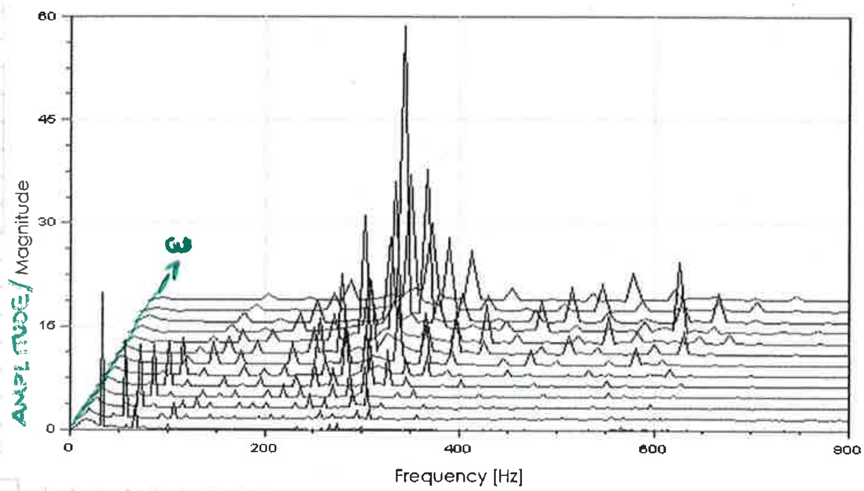
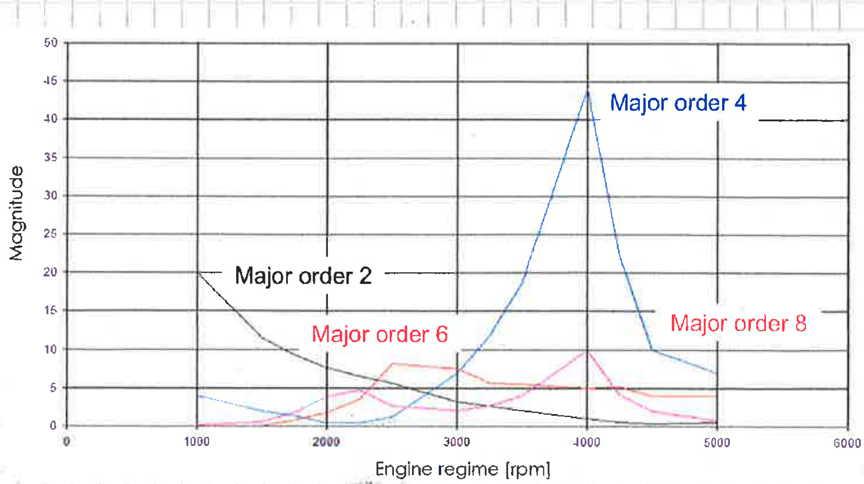
- THE WATERFALL DIAGRAM (3D) AND THE ORDER CHART (2D) REPRESENTATION CAN BE OBTAINED BY POST-PROCESSING THE MB SIMULATION RESULTS.
- THE ANGULAR VELOCITY OF THE PULLEY CAN BE CHOSEN AS REFERENCE VARIABLE BECAUSE IT ALLOWS TO IDENTIFY THE PEAKS CORRESPONDING TO THE NATURAL FREQUENCIES OF THE TORSIONAL MODES WITH THE SEISMIC MASS IN PHASE OR OUT OF PHASE WITH THE CRANKSHAFT ROTATION.
- THESE DIAGRAMS REPRESENT THE "DYNAMIC IDENTITY CARD" OF THE CRANKSHAFT AND THEY ARE USEFUL TO IDENTIFY ITS CRITICAL REGIMES:
  - DEPENDING ON THE OPERATING CONDITION, THE CRANKSHAFT RESPONDS TO THE EXTERNAL LOADS IN DIFFERENT WAY.
  - THERE ARE REGIMES IN WHICH THE CRANKSHAFT HAS A RESONANT RESPONSE
  - AT THESE REGIMES IT IS NECESSARY TO VERIFY THE STRESS STATE OF THE CRANKSHAFT IN ORDER TO EVALUATE ITS MOST CRITICAL CONDITION.



THE WATERFALL DIAGRAM IN PLANE ω-λ CORRESPONDS TO THE CAMPBELL DIAGRAM



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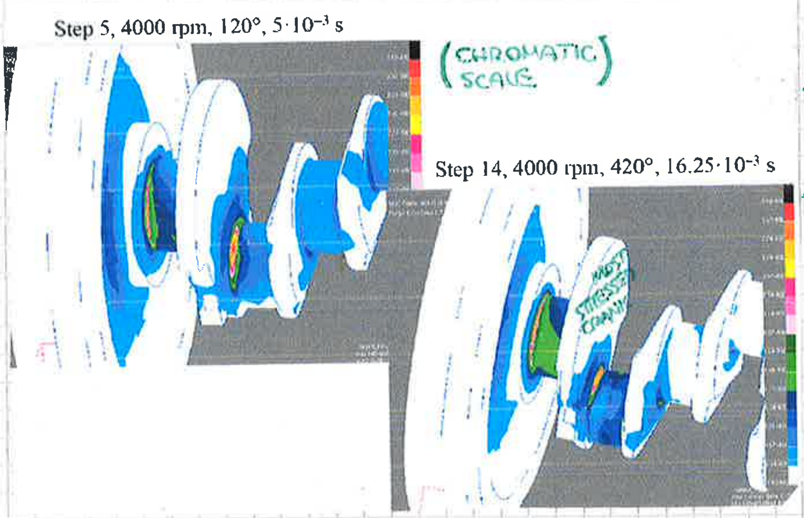
- BY HAVING AVAILABLE THE SIMULATION RESULTS FOR EACH ENGINE REGIME, THE **NEXT STEP** IS THE EVALUATION OF THE **STRESS STATE** OF THE CRANKSHAFT.
- AS THE **LOADING CONDITION** OF THE CRANKSHAFT IS **DIFFERENT AT EVERY TIME INSTANT**, IT IS NECESSARY TO **FOCUS THE ATTENTION** ON **THOSE LOAD CONDITIONS** THAT ARE **PARTICULARLY CRITICAL** FOR THE CRANKSHAFT RESISTANCE.
- FROM THE **ORDER CHART**, THE **MOST CRITICAL ENGINE SPIN SPEED** CAN BE IDENTIFIED AND THE **CORRESPONDING STRESS STATE** OF THE CRANKSHAFT CAN BE COMPUTED.
- AT THIS POINT OF THE NUMERICAL ANALYSIS, WITH A DUAL OPERATION WITH RESPECT TO THE DYNAMIC CONDENSATION OF DOFS, **THE OUTPUT RESULTS (DISPLACEMENTS AT THE MASTER NODES)** OF THE MB MODEL ARE **REDIRECTED AS INPUT DATA OF THE FE MODEL** OF THE CRANKSHAFT. (MBA → FEA)
- WITH THIS DATA RECOVERY PROCESS AND **BY DIVIDING THE ENGINE CYCLE** IN A **DISCRETE NUMBER OF STEPS**, THE **COMPUTATION OF THE STRESS STATE** CAN BE MADE **AT A CERTAIN NUMBER OF DEGREES** OF THE CRANKSHAFT ROTATION, IN ORDER TO **IDENTIFY THE MOST STRESSED PART** OF THE CRANKSHAFT.



MOST STRESSED PARTS:
- RADIUS AT E. FLYWHEEL-MAIN JOURNAL INTERFACE.
- RADIUS AT AT MAIN J.-CHEEK INT. NEAREST TO THE FLYWHEEL. (BECAUSE OF ITS HIGH GYROSCOPIC EFFECT).

FIRST WE DID: **FEA → MBA** (OVERALL DOFS → MASTER DOFS)
NOW WE HAVE TO DO THE DUAL OPERATION: **MBA → FEA** (MASTER DOFS → OVERALL DOFS)
[**DATA RECOVERY PROCESS**: WE REACTIVATES THE SLAVE DOFS]

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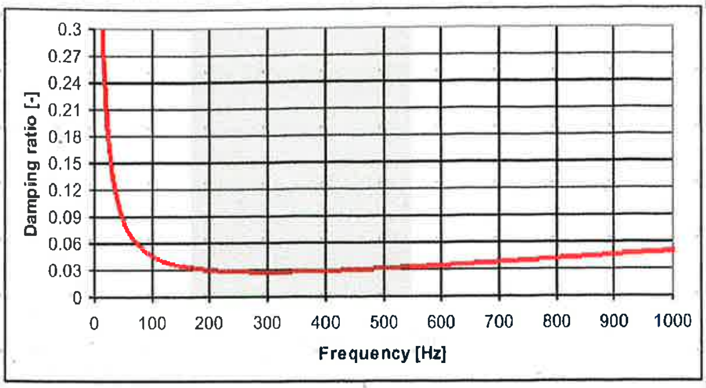
FLEXIBLE BODIES: IN THE FE-MB MODEL **CRANKSHAFT**, **FLYWHEEL** AND **CYLINDER BLOCK** WERE CONSIDERED **FLEXIBLE**.

DAMPING: IN THE FREQUENCY RANGE OF INTEREST (180-550 [Hz]), A CRITICAL DAMPING OF 3% WAS CONSIDERED, ACCORDING TO THE RAYLEIGH CURVE:



JOINTS: IN THE FE-MB MODEL, DIFFERENT LINK ARE CHOSEN: **KINEMATIC LINK**, **2D HYDRODYNAMIC BEARING**, AND **3D HYDRO-DYNAMIC BEARING** (MORE ACCURATE, TAKES INTO ACCOUNT THE MOMENTS ACTING ON THE BEARINGS).

Component		Connection type	
Pin	Piston	Kinematic link	
Pin	Connecting rod	Kinematic link	
Piston	Liner	Kinematic link	
Connecting rod	Crankshaft	2D hydrodynamic bearing	CRANKPIN BEARING
Crankshaft	Block	3D hydrodynamic bearing	MAIN BEARING
Block	Ground	Kinematic link	
Pulley	Crankshaft	Kinematic link	

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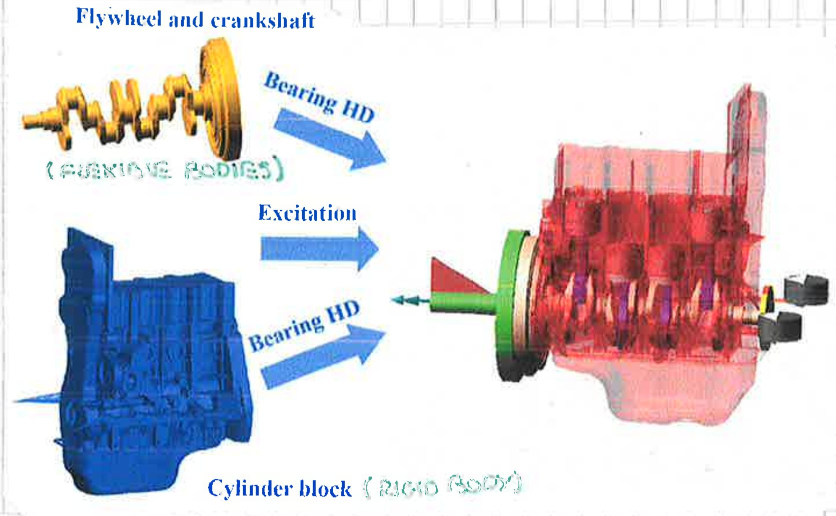
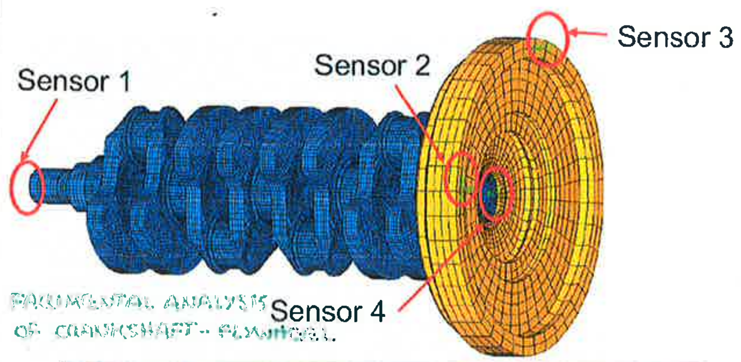


AFTER CONSIDERING THE PRINCIPAL AND SECONDARY EXCITATIONS, WE ASSEMBLE ALL THE BODY ELEMENT CONSIDERING THE HYDRO-DYNAMIC BEARINGS CONNECTIONS.

- DNA - SENSOR POINTS FOR THE MB MODEL RESULTS



WE IDENTIFY SOME **KEY POINTS** (SENSOR 1,2,3,4) OF MBr TO REPRESENT THE DYNAMIC BEHAVIOUR OF THE MB MODEL. ALL THESE **KEY NODES** ARE USEFUL TO IDENTIFY A **SPECIFIC BEHAVIOUR**

Sensor	Transcription
1-4	Torsion of crankshaft
3-4	Relative axial displacement of flywheel
2-4	Relative axial displacement of flywheel
2	Absolute axial displacement of flywheel
3	Absolute axial displacement of flywheel
4	Irregular motion

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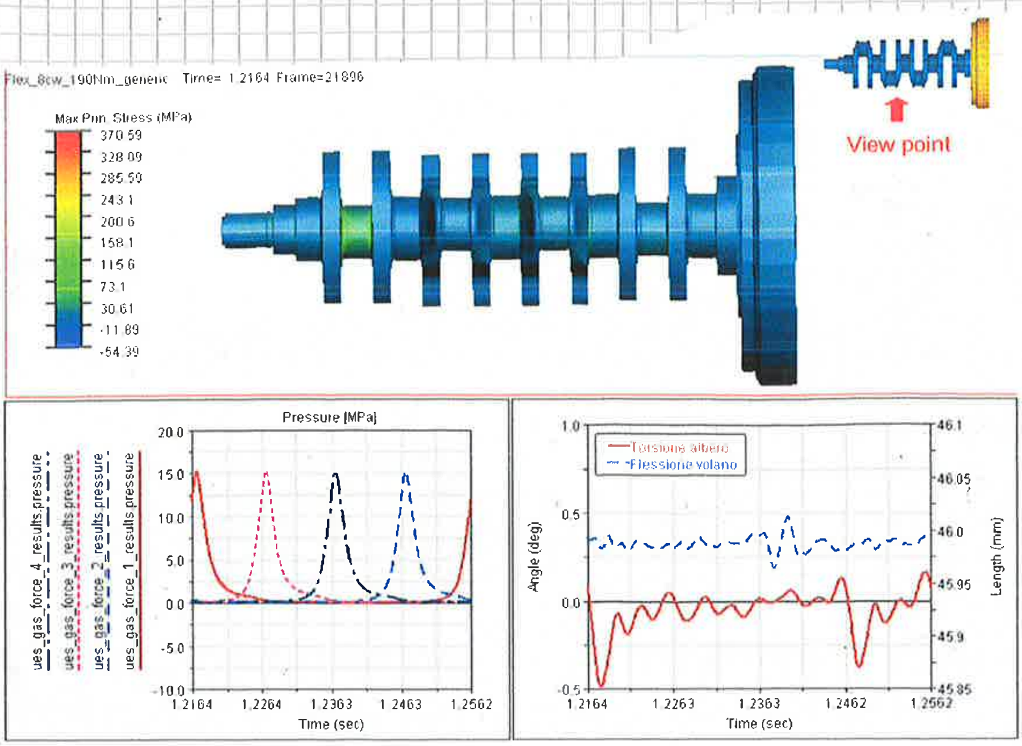
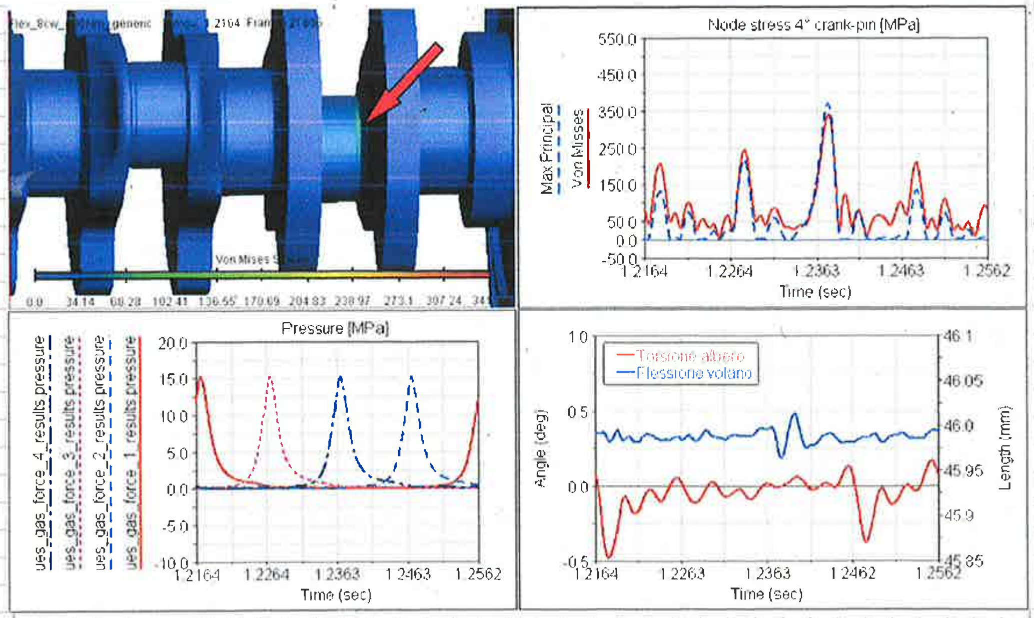


WE OBTAIN THE RESULTS OF THE FALIMENTAL AN., IN TERMS OF DISPLAC.S OF C.S. AND E.F.



FINALLY, THROUGH DATA RECOVERY P. WE OBTAIN THE STRESS STATE IN THE MOST CRITICAL SECTION.

FROM MSD:

7th MODE: 1° BENDING (XY); 277 [Hz]

11th MODE: 1ST TORSION; 765 [Hz]

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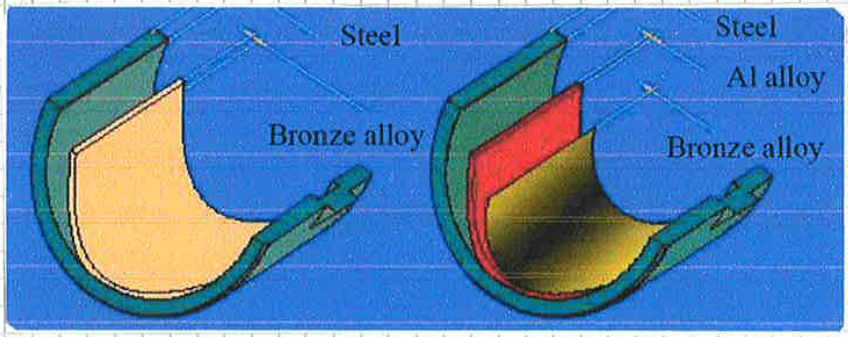
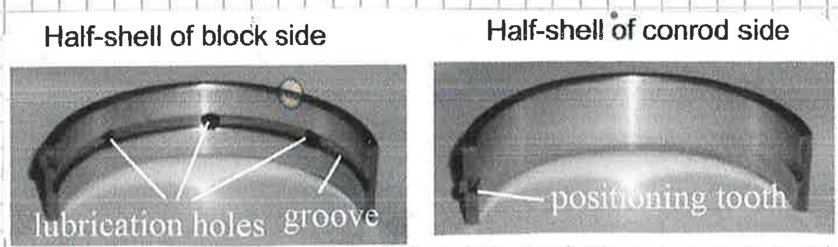
• ARCHITECTURE AND GEOMETRY

- THE CRANKSHAFT JOURNALS ROTATE ON TWO HALF-SHELLS (BUSHINGS OR BEARINGS) INSERTED INTO THE MAIN CAPS AND THE WALLS OF THE CYLINDER BLOCK AND INTO THE BIG EYE OF THE CONNECTING ROD.
- EACH HALFSHELL IS MADE OF ONE STEEL LAYER, THAT IS THE STRUCTURAL PART OF THE BEARING, AND ONE ADDITIONAL LAYER (BI-METALLIC SOLUTION) OR TWO ADDITIONAL LAYERS (TRI-METALLIC SOLUTION) OF ANTI-FRICTION MATERIALS THAT:
  - CREATE A BEARING SURFACE TO SUPPORT THE LOADS GENERATED BY THE CRANK MECHANISM AND TRASMITTED THROUGH THE BEARING OIL FILM.
  - GUARANTEE AN ADEGUATE WEAR RESISTANCE AT LOW SPIN SPEED, WHEN THE CONTACT IS OF UNCTUOUS TYPE.
  - ABSORB THE GEOMETRY INACCURACIES OF THE JOURNALS.



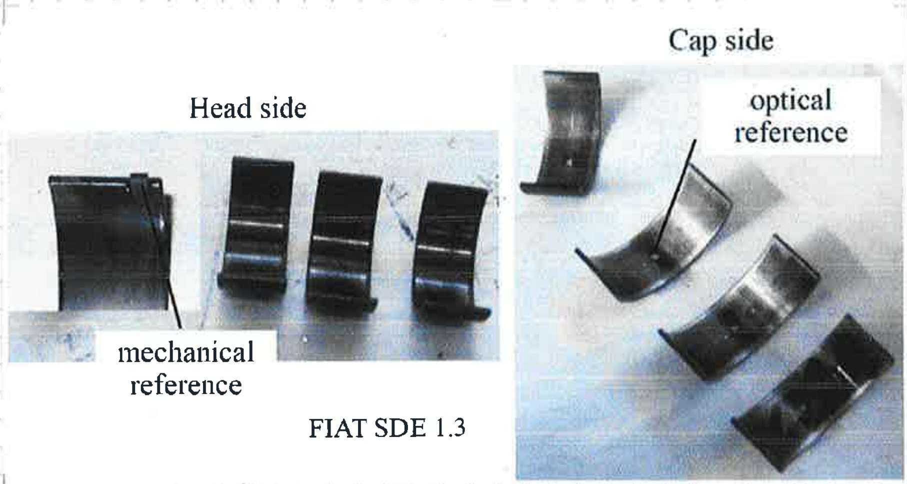
+ BRONZE ALLOY (BI-METALLIC) OR + BRONZE ALLOY + Al-ALLOY (TRI-METALLIC)

- IN GENERAL, ONE OF THE TWO HALF-SHELLS PRESENTS A CENTRAL GROOVE WITH RADIAL HOLES TO COLLECT THE OIL (THAT COMES FROM CHANNELS PRESENT IN THE CYLINDER BLOCK) CONTINUOSLY FEEDING THE CRANKSHAFT AND LUBRICATING THE CONNECTING RODS. (HALF-SHELL SUPPORTED BY THE CAP, SO IT'S THE LOWER PART)
- BECAUSE THE HALF-SHELLS INSERTED INTO THE MAIN CAPS OF THE CYLINDER BLOCK ARE HAVILY LOADED, IN ORDER TO GUARANTEE AN ADEGUATE LOADING SURFACE, IN DIESEL ENGINES THE CENTRAL GROOVE IS ELIMINATED AND THE MAIN JOURNALS PRESENT A DOUBLE DIAGONAL DRILLING SOLUTION.
- TO DISTINGUISH THE TWO HALF-SHELLS, A MECHANICAL REFERENCE (POSITIONING TOOTH) IS PRESENT ON THE HALF-SHELL OF THE "CAP SIDE".



- FURTHERMORE, TO AVOID ANY POSSIBLE DAMAGE TO THE BUSHING MATERIAL, A TAPERED SIDE IS PRESENT IN THE CONTACT AREA BETWEEN THE TWO HALF-SHELLS.
- IN THIS WAY, THE COMPRESSION FORCE DEFORMS THE TWO IN CONTACT TAPERED ENDS OF THE HALF-SHELLS WITHOUT CREATING ANY INSTABILITY ACTION.
- THE ADHERENCE TO THE BIG EYE SURFACE IS THEN ASSURED AND THE CIRCUMFERENTIAL PROFILE OF THE BUSHING SHOULD BE SIGHTLY HIGHER THAN THE CIRCUMFERENTIAL PROFILE OF THE CONROD BIG EYE.
- FOR MEDIUM STRESSED ENGINES, TWO HALF-SHELLS OF THE SAME TYPE FOR BOTH THE HEAD SIDE AND THE CAP SIDE ARE USED.
- FOR HIGH STRESSED ENGINES, THE HALF-SHELL ON THE CAP SIDE IS MADE OF STEEL WITH HIGHER MECHANICAL CHARACTERISTICS WHILE THAT ON THE HEAD SIDE IS MADE OF STANDARD STEEL.
- IN THIS CASE, TO AVOID INCORRECT ASSEMBLY, DIFFERENT REFERENCE MARKS ARE ADOPTED.

- MECHANICAL REFERENCES: TOOTH OR RIB ON THE HALF-SHELL (GROOVES ON THE HEAD)
- OPTICAL REFERENCES: HOLE ON THE HALF-SHELL OF THE CAP SIDE.



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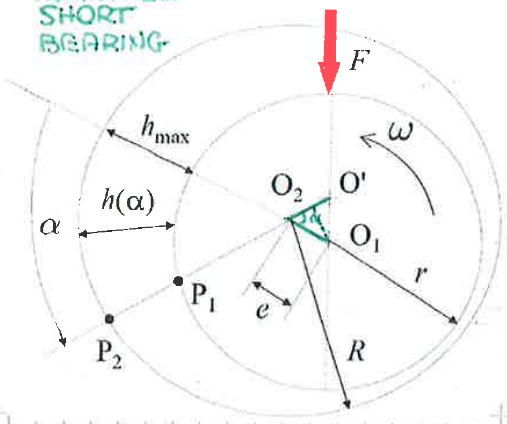
INFINITELY SHORT BEARING



- RADIAL CLEARANCE (δ) IS DEFINED AS THE DIFFERENCE BETWEEN BEARING RADIUS (R) AND MAIN JOURNAL RADIUS (R):

  $\delta = R - R$ ($g = 2\delta$ DIAMETRAL CLE/GAP)

- ECCENTRICITY (e) BETWEEN MAIN JOURNAL AND BEARING IS DEFINED AS THE DISTANCE BETWEEN CENTER (O_2) AND MAIN JOURNAL CENTER (O_1):

  $e = \overline{O_2 O_1}$

- THE OIL FILM THICKNESS ($h(\alpha)$) HAS A CONVERGENT-DIVERGENT PROFILE.
- THE ANGULAR COORDINATE (α) IS ASSUMED FROM THE BEARING POINT CORRESPONDING TO THE MAXIMUM OIL FILM THICKNESS (h_{MAX}).

- THE OIL FILM THICKNESS IS:

$$h(\alpha) = \overline{P_1P_2} = \overline{O_2P_2} - \overline{O_2P_1} = R - (\overline{O'P_1} - \overline{O'O_2})$$

BEING $\overline{O'P_1} \approx R$, $\overline{O'O_2} \approx e \cdot \cos\alpha$

$$h(\alpha) = R - R - (-e \cdot \cos\alpha) = \delta + e\cos\alpha = \delta(1 + \varepsilon\cos\alpha) \quad , \quad \varepsilon = e/\delta \longleftrightarrow e = \varepsilon\delta$$

WHERE $\varepsilon = \frac{e}{\delta}$ IS THE RATIO BETWEEN THE BEARING ECCENTRICITY (e) AND THE RADIAL CLEARANCE (δ).

- THE SOLUTION OF THE HYDRODYNAMIC PROBLEM (IN TERMS OF OIL FILM PRESSURE P) CAN BE OBTAINED BY INTEGRATING THE REYNOLDS EQUATION (1D)

$$\frac{dp}{d\alpha} = \frac{6R\mu v}{h^2(\alpha)}\left[1 - \frac{2q}{vh(\alpha)}\right]$$

(INFINITELY SHORT BEARING → MONODIMENSIONAL REYNOLD EQ.)
SOLVING THIS WE FIND THE P-TREND IN FUNCTION OF ANG. COORD. α

WITH:

P, q : OIL FILM PRESSURE AND OIL FLOW RATE

μ : OIL DYNAMIC VISCOSITY

v : RELATIVE VELOCITY BETWEEN THE TWO LUBRICATED AND COUPLED SURFACES

hp1
- THE ANALYTICAL SOLUTION OF THE REYNOLD EQUATION IS POSSIBLE UNDER THE ASSUMPTION OF NULL PRESSURE AT THE POINT OF MAXIMUM THICKNESS (h_{MAX}) (I.E. MAXIMUM MEATUS) OF THE OIL FILM.

hp2
- THIS CONDITION IS POSSIBLE ONLY IF THE BEARING OIL SUPPLY OCCURS AT THE SAME ANGULAR POSITION OF THE MAXIMUM MEATUS (BUT DEVIATIONS OF ± 20° ARE HOWEVER ACCEPTED, AND THE OIL SUPPLY POSITION CAN BE DEFINED).

IF THIS SIMPLIFIED hps ARE NOT GUARANTED THEN WE HAVE TO SOLVE REYNOLDS EQ. USING A NUMERICAL APPROACH → SOLUTION BY NEWTON-RAPHSON OR RUNGE-KUTTA METHODOLOGIES (TO SOLVE THIS MONODIMENSIONAL EQ.).

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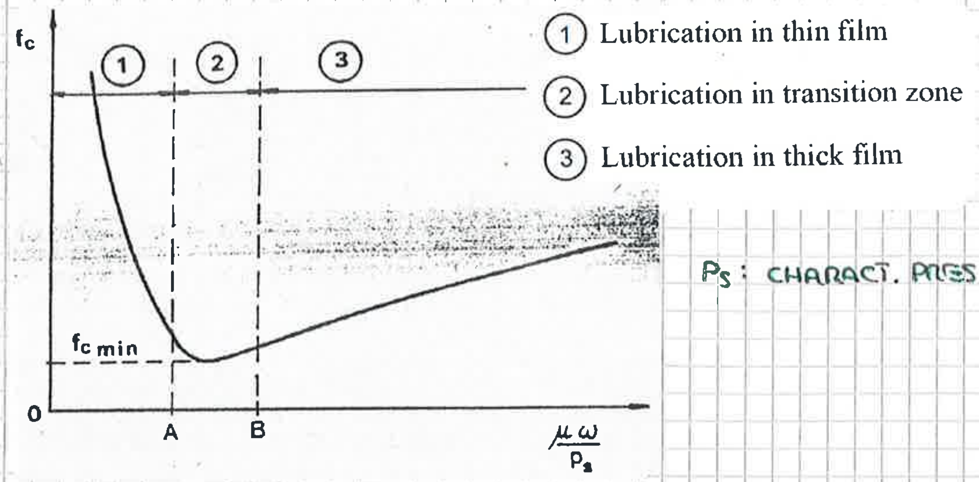
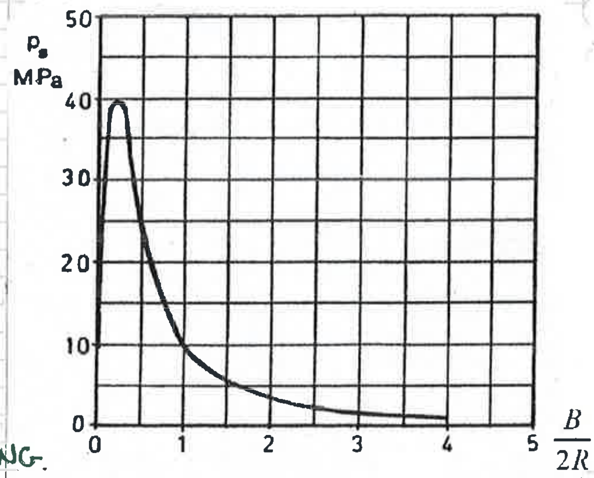
- THE FRICTION COEFF. (f_c) DEPENDS ON THE OIL DYNAMIC VISCOSITY (μ), THE CHARACTERISTIC PRESSURE (P_S) AND THE ANGULAR VELOCITY (ω).



P_S: CHARACT. PRESSURE / LOAD CAPACITY

- THE LOAD CAPACITY OF THE BEARING DEPENDS ON THE ELONGATION PARAMETER.
- FOR LOW VALUES OF ELONGATION PARAMETER ($B/2R$), THE CHARACTERISTIC PRESSURE (P_S) DECREASES AND THE OIL CAN LEAKS FROM THE BEARING ENDS CREATING NON-CORRECT CONDITIONS FOR HYDRODYNAMIC LUBRICATION.
- FOR HIGHER VALUES OF ELONGATION PARAMETER ($B/2R$), THE CHARACTERISTIC PRESSURE (P_S) SLOWLY DECREASES DUE TO POSSIBLE MISALIGNMENT BETWEEN JOURNAL AND BEARING.



- INCREASING THE LOAD (F) APPLIED TO THE COUPLING JOURNAL-BEARING OR DECREASING THE ENGINE SPIN SPEED (ω), THE MINIMUM OIL FILM THICKNESS (h_0) DECREASES.
- THE MINIMUM OIL FILM THICKNESS MUST BE AT LEAST EQUAL TO THE SUM OF THE PEAK-VALLEY ROUGHNESS OF THE COUPLED SURFACES JOURNAL-BEARING AND IN GENERAL IT IS ASSUMED EQUAL TO:

$h_0 \geq (3.5 \cdot R_a + 25 R)$, R_a AND h_0 IN $[\mu m]$, R IN $[m]$

(SEMI-EMPIRICAL FORMULA ⇒ WE CAN USE DIFF. UNITS OF MEASURE)

- **BEARING LUBRICATION SOLUTION**

- THE HYDRODYNAMIC PRESSURE GENERATED AT MAIN BEARINGS AND CRANKSHAFT JOURNALS INTERFACE, AS WELL AS CONNECTING ROD BIG-END BEARINGS, IS GOVERNED BY 2D REYNOLD EQUATION:

$$\frac{\partial}{R_j^2 \partial\alpha}\left(h^3\frac{\partial P}{\partial\alpha}\right)+\frac{\partial}{\partial x}\left(h^3\frac{\partial P}{\partial x}\right)=6\eta\frac{U\partial h}{R_j\partial\alpha}+12\eta\frac{\partial h}{\partial t}$$

, η : KINEMATIC VISCOSITY

WHERE THE OIL FILM THICKNESS (h) AT JOURNAL-BEARING INTERFACE CAN BE DEFINED AS: $(h(\alpha)=\delta(1+\varepsilon\cos\alpha))$ $(\varepsilon=e/\delta)$

$h(x,\alpha)=c(1+\varepsilon\cos\alpha)$, c: RADIAL CLEARANCE FUNCTION OF AXIAL COORDINATE $(c=c(x))$

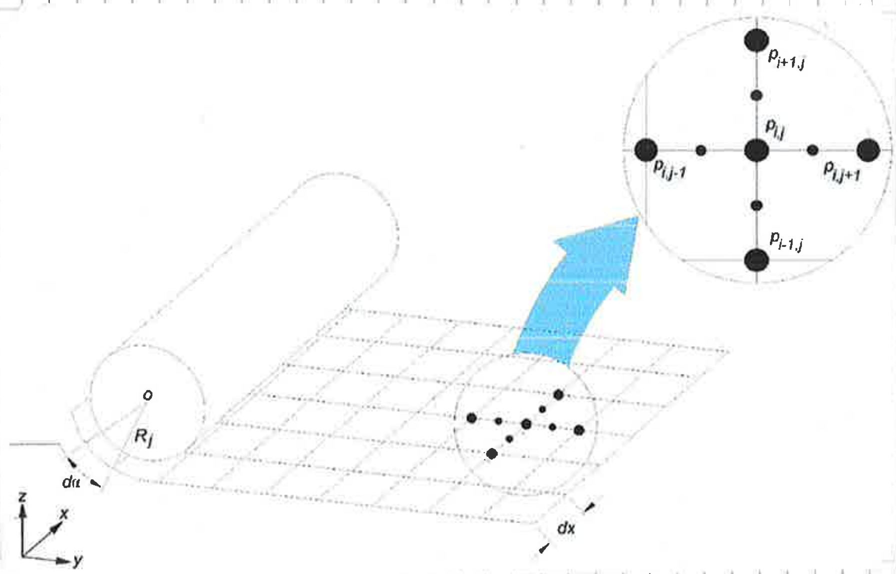
WITH THE ECCENTRICITY RATIO (ε) BETWEEN BEARING BODIES AXES TO CLEARANCE:

$$\varepsilon=\frac{e}{c}$$

AND THE SPEED OF ENTRAINING MOTION CAN BE CALCULATED AS:

$$U=\frac{1}{2}\omega R_j \qquad (U \text{ or } V)$$

- FINITE DIFFERENCE SCHEME CAN BE APPLIED TO SOLVE 2D REYNOLDS EQUATION.
- A FINITE DIFFERENCE MESH CAN BE DEFINED OVER THE JOURNAL.



- APPLYING THE FINITE DIFFERENCE SCHEME THE 2D REYNOLDS DIFFERENTIAL EQUATION REDUCES TO AN ALGEBRAIC EQUATION.

$$\frac{h^3_{i+1/2,j}\frac{P_{i+1,j}-P_{i,j}}{\Delta x}-h^3_{i-1/2,j}\frac{P_{i,j}-P_{i-1,j}}{\Delta x}}{\Delta x}+\frac{1}{R_j^2}\left(\frac{h^3_{i,j+1/2}\frac{P_{i,j+1}-P_{i,j}}{\Delta\alpha}-h^3_{i,j-1/2}\frac{P_{i,j}-P_{i,j-1}}{\Delta\alpha}}{\Delta\alpha}\right)=$$

$$6\eta U\frac{h_{i+1/2,j}-h_{i-1/2,j}}{\Delta x}+12\eta\omega\frac{h^{\theta+\Delta\theta}_{i,j}-h^{\theta}_{i,j}}{\Delta\theta} \qquad \frac{1}{R_j\Delta\alpha}=\frac{1}{\Delta x}$$

- SOLVING THIS ALGEBRAIC EQUATION IT IS POSSIBLE TO COMPUTE THE PRESSURE DISTRIBUTION AT JOURNAL-BEARING INTERFACE.
- KNOWING THE PRESSURE DISTRIBUTION WE CAN COMPUTE THE MINIMUM THICKNESS (h_0).
- TO SOLVE THE PREVIOUS ALGEBRAIC EQ. WE NEED THE B.C.

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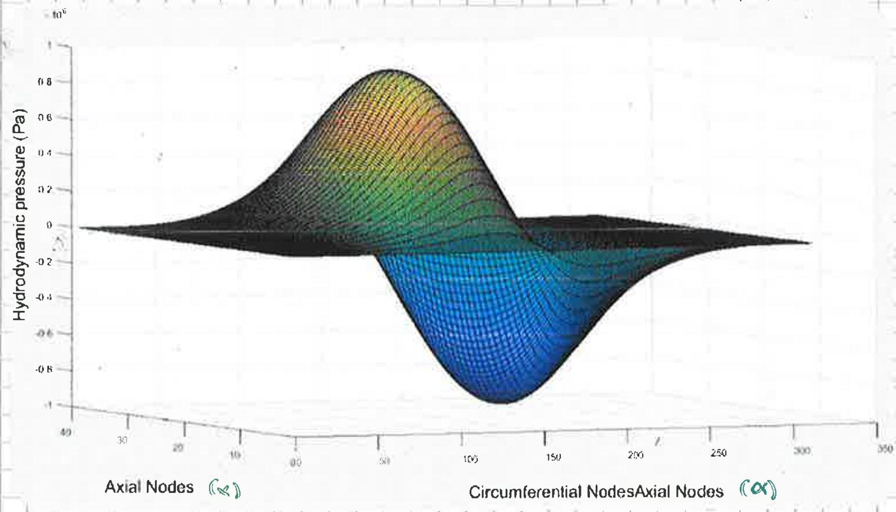
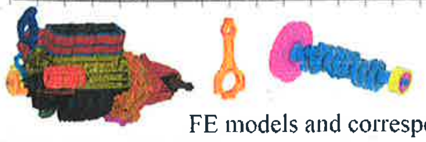
- AFTER SOLVING THE FIRST ORDER DIFFERENTIAL EQUATION OF FORCE BALANCE, THE INSTANTANEOUS ECCENTRICITY BETWEEN JOURNAL AND BEARING CAN BE DEFINED TO UPDATE OIL FILM THICKNESS AT EACH CRANK ANGLE TO CALCULATE HYDRODYNAMIC PRESSURE (P_h) AT BEARING SURFACES INTERFACE.

($e = \overline{O_1 O_2}$: ECCENTRICITY)

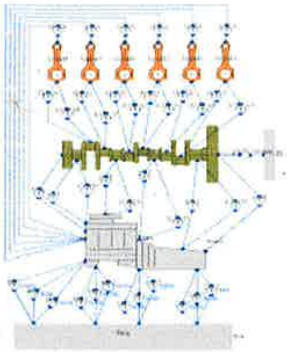


RESOLVING THE DIMESIONLESS FORM OF 2D REYNOLDS EQ. → HYDROD. P DISTRIBUTION (P_h) → RADIAL FORCE (F_R) AND TANGENTIAL FORCE (F_t) → $F_h = \sqrt{F_R^2 + F_t^2}$: HYDROD. FORCE (F_h) (NOT SO CLEAR ...)

- NC FEA-MBA - PROCEDURE



FE models and corresponding dynamic DoF reduction



MB model definition

- Elasto-hydrodynamic solution
- 3D distribution of oil film pressure and thickness
- Oil flow rate computation

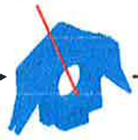
- NC FEA-MBA - MODEL DEFINITION

Detailed bearing geometry →

→ Holes for oil supply
→ Oil distribution groove
→ Half-shells junction
→ Roughness of coupled parts

Bearing load application

Elasto-hydrodynamic lubrication →



→ Numerical solution of Reynolds Equation
Model coupling → Update of the oil film deformed configuration
→ Structural solution (bearing deformation)

TAKES INTO ACCOUNT :

Oil characteristics → Viscosity depends on temperature and pressure →

3D distribution of the oil film pressure and thickness

- NC FEA-MBA - DIFFERENT MODEL COMPLEXITY

COMPLEXITY ↓

1st STAGE OF COMPLEXITY

CR MODEL
Rigid main journal and rigid bearing
NO misalignments
NO crankshaft dynamics

→ All rigid, no FE models are required (only nonlinear spring elements)

2nd

CE MODEL
Rigid main journal and elastic bearing
NO misalignments
NO crankshaft dynamics

→ FE model of single wall and cap

3rd

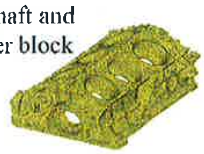
BR MODEL
Elastic crankshaft and rigid cylinder block
YES misalignments
YES crankshaft dynamics

→ FE model of crankshaft

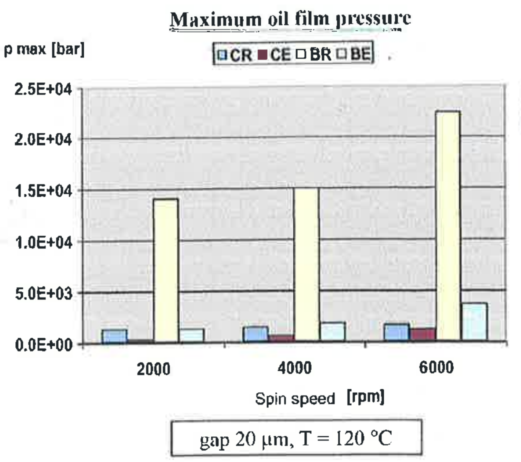
4th

BE MODEL
Elastic crankshaft and elastic cylinder block
YES misalignments
YES crankshaft dynamics
YES cylinder block dynamics

→ FE model of crankshaft and cylinder block

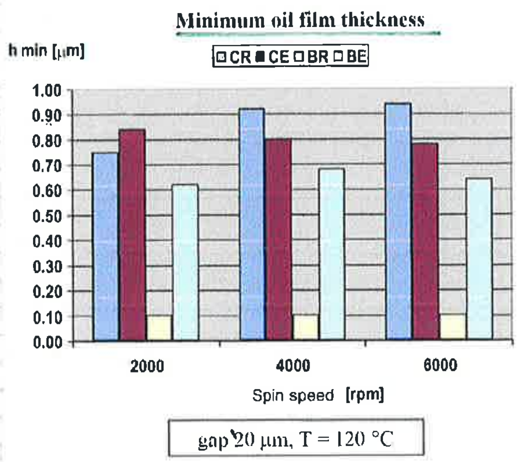


CE : ELASTIC BEARING MODEL
BE : ELASTIC CYLINDER BLOCK



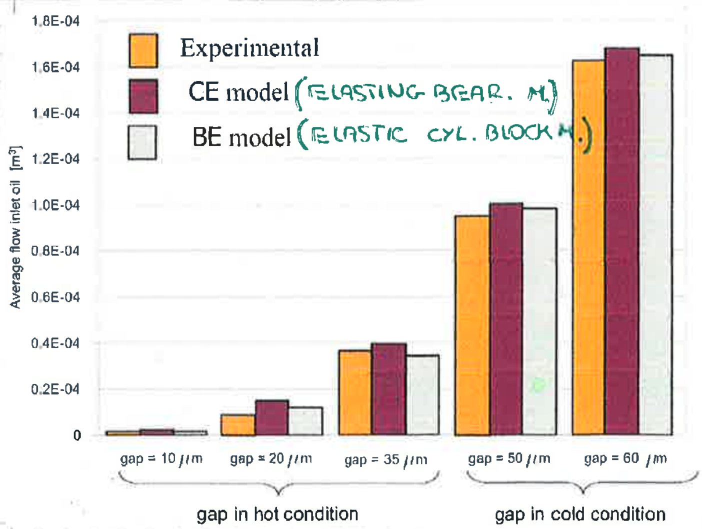
Models CR CE BE provide similar values and trends

Model BR is not adequate because:
- Excessive over-estimation of the oil film pressure (1 order of magnitude)
- Wrong seizure condition evaluation
- Misalignments over-estimation



Models CR CE BE provide similar values and trends

Model BR is not adequate because:
- Excessive under-estimation of the oil film thickness (1 order of magnitude)
- Wrong seizure condition evaluation
- Misalignments over-estimation



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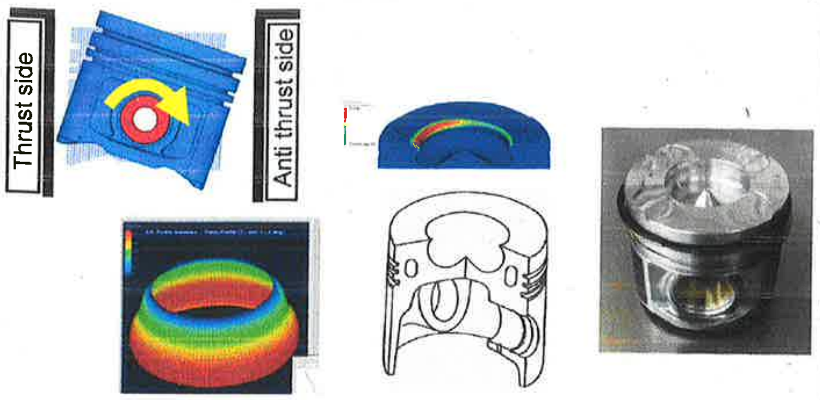
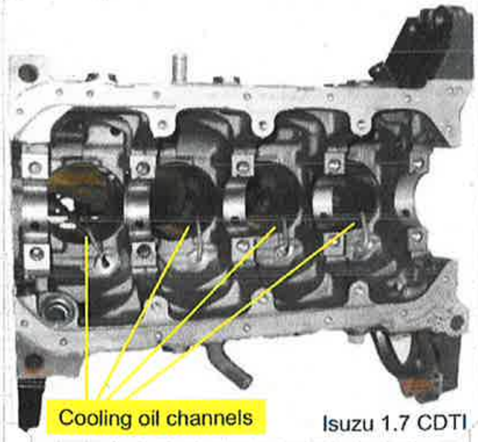
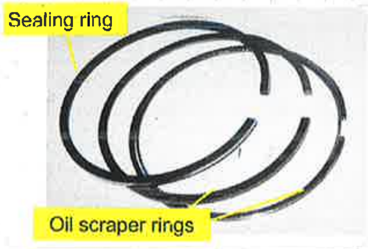
E6 PISTON



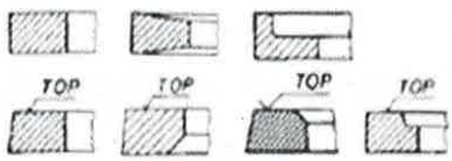
- ARCHITECTURE AND GEOMETRY
- MATERIALS
- DESIGN GUIDELINES – PISTON
- DESIGN GUIDELINES – PISTON RINGS
- DESIGN AND VERIFICATION GUIDELINES – PISTON RINGS
- NUMERICAL ANALYSIS – PISTON RINGS
- PISTON RING LUBRICATION
- FRICTION FORCE
- NUMERICAL ANALYSIS – PISTON
- NUMERICAL ANALYSIS – PISTON STATIC THERMAL ANALYSIS
- NUMERICAL ANALYSIS – PISTON THERMO-STRUCTURAL ANALYSIS
- PISTON SLAP
- PISTON SLAP – SIMPLIFIED STUDY WITH FREE BODY DIAGRAMS
- PISTON SKIRT LUBRICATION
- PISTON SLAP – EFFECTS
- METHODS TO DECREASE PISTON SLAP
- NUMERICAL FEA-MBA OF PISTON SLAP



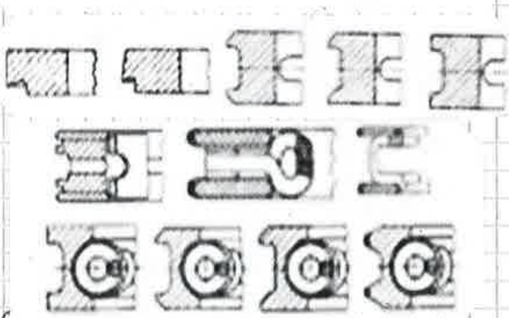


- PISTON RINGS ENSURE TIGHTNESS OF THE COMBUSTION CHAMBER AND MINIMIZE GAS/OIL PASSAGE TOWARDS THE CRANKCASE (BLOW-BY PHENOMENON) / TOWARDS THE COMBUSTION CHAMBER.
- THEY CONTROL THE OIL AMOUNT THAT LUBRICATES THE LINER INTERNAL SURFACE AND ENSURE A PROPER LUBRICATION TO AVOID SLIDING AND WEAR.
- IN GENERAL THERE ARE 3 RINGS FOR EACH PISTON: THE FIRST RING SEALS THE GAS (SEALING RING) THE SECOND AND THIRD RINGS CONTROL THE OIL AMOUNT (OIL SCRAPER RINGS).
- DEPENDING ON THE TASK, EACH RING HAS A CROSS SECTION WITH A SHAPE THAT DEFINES THE TYPE OF SUPPORT OF THE LINER (RADIAL OR ALONG THE EDGE) AND THE CORRESPONDING LOAD APPLIED IN RADIAL AND TANGENTIAL DIRECTION.

- SEALING RING CROSS SECTIONS: FIRST ROW PROFILES WITH "RADIAL LINER SUPPORT", SECOND ROW PROFILES WITH "EDGE LINER SUPPORT".

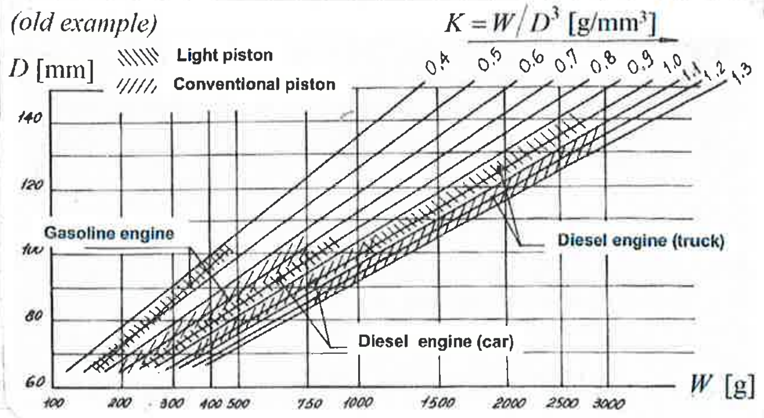


- OIL SCRAPER RING CROSS SECTION:
- OIL SCRAPER RING (THIRD GROOVE) CAN ALSO ACCOMODATE AN INTERNAL SPIRAL SPRING TO INCREASE THE TANGENTIAL LOAD ON THE LINER
- IN SOME CASES, OIL SCRAPER RING CAN BE ALSO SUBDIVIDED IN A CIRCUMFERENTIAL SPRING AND TWO RADIAL PARTS THAT DIRECTLY SUPPORT THE LINER.



• DESIGN GUIDELINES – PISTON

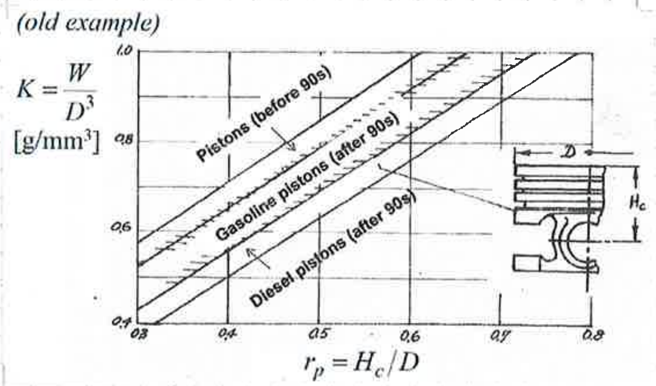
- FROM THE VALUE OF THE BORE DIAMETER (D) AND THE ESTIMATED WEIGHT (W) OF THE PISTON, THE SO-CALLED APPARENT DENSITY (K) CAN BE EVALUATED BY USING SEMI-EMPIRICAL GRAPHS WHERE ENGINES ARE CLASSIFIED AS "GASOLINE" AND "DIESEL" (CAR AND TRUCK) AND PISTONS AS "CONVENTIONAL" AND "LIGHT".



SEMI-EMPIRICAL <-> NOT ANALYT. NOR NUM

$D, W \rightarrow K = \frac{W}{D^3}$: APPARENT DENS.

- BY INTRODUCING THE APPARENT DENSITY (K) IN ANOTHER SEMI-EMPIRICAL GRAPH, THE SO-CALLED GEOMETRICAL RATIO (R_p) OF THE PISTON CAN BE OBTAINED:



$K = \frac{W}{D^3} \rightarrow R_p = \frac{H_c}{D}$: GEOMETR. RATIO

$D, R_p \rightarrow H_c$: COMPRESSION HEIGHT

- THEN THE COMPRESSION HEIGHT (H_c) BETWEEN PISTON TOP AND PIN AXIS CAN BE COMPUTED.

- BY REFERRING TO THE ALLOWABLE PRESSURE FOR THE PISTON MATERIAL:

Piston material	Allowable pressure p_{adm} [MPa]
Al alloy	0.98
Cast iron	0.44 – 0.64

- THE LENGTH OF THE SKIRT CAN BE COMPUTED AS:

$$P_{ADM} = \frac{F_{n,MAX}}{L \cdot D} \Rightarrow L = \frac{F_{n,MAX}}{P_{ADM} \cdot D} \quad : \text{ LENGHT OF SKIRT}$$

- ANOTHER WAY TO DEFINE THE FIRST ATTEMPT LENGTH OF THE SKIRT IS TO CONSIDER A UNIFORM PRESSURE DISTRIBUTION; [AS THE PIN AXIS SHOULD BE AT HALF OF THE PISTON COMPRESSION HEIGHT (H_c)] THEN: ?

$L = 2H_c$ (SEMI-EMPIRICAL FORMULA)

- DURING ENGINE OPERATING CONDITION, THE TEMPERATURES REACHED IN THE PISTON TOP ARE HIGHER THEN THOSE REACHED IN THE PISTON SKIRT.
- BECAUSE MOST OF THE PISTON MATERIAL IS LOCATED IN THE AREA BETWEEN THE PIN AXIS AND THE PISTON TOP, THE PISTON SUFFERS OF AN HIGH HEAT DISTORSION.
- PISTON IS THEN MADE WITH A CONICAL SHAPE SO THAT, WHEN IT IS IN OPERATING CONDITION IT ASSUMES ITS OPTICAL CYLINDRICAL SHAPE AVOIDING THE RISK OF SEIZURE IN THE LINER.
- THE CONTACT PISTON-LINER IS GUARANTED BY ELASTIC RINGS WHOSE POSITION ALLOWS ALSO TO CONTROL THE HEAT FLOW THROUGH THE PISTON
- THE GAP BETWEEN THE SKIRT AND THE LINER IS DEFINED WITH RESPECT TO THE LINER MATERIAL.

Liner material	Gap piston skirt-liner [mm]
Al alloy	0.04 – 0.06
Cast iron	0.09 – 0.11

(HYPER-EUTECTIC AL-ALLOY PRECISELY)

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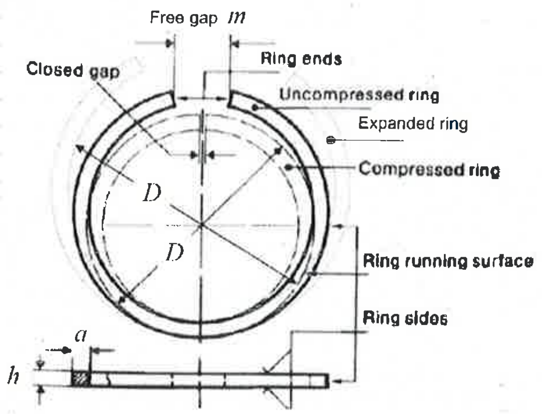
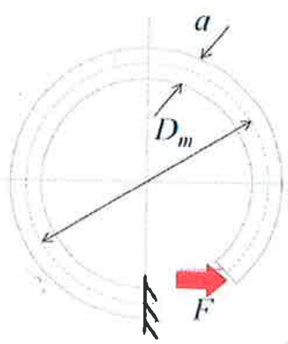




WE ARE EVALUATING THE STRUCTURAL INTEGRITY OF THE PISTON RING.

- THE PISTON RING IS MODELLED AS A CURVED BEAM FIXED AT ONE END AND LOADED AT THE OTHER END BY A CONCENTRATED FORCE THAT IS FUNCTION OF THE MAXIMUM ADMISSIBLE PRESSURE OF THE PISTON RING ON THE LINER.

$F = \frac{P_{ADM} D_m h}{2}$, $D_m = D + a$: MEAN DIAMETER BETWEEN THE D OF EXP. RING AND D OF COMP. R.

- THE ADMISSIBLE PRESSURE (P_{ADM}) IS A COMPROMISE BETWEEN THE WEAR MINIMIZATION ON THE CONTACT AREA LINER - PISTON RING AND THE REQUIRED SEALING ACTION; ITS VALUE DEPENDS ON THE BORE (D).

D [mm]	p_{adm} [MPa]
$180 \le D \le 400$	0.055 – 0.06
$410 \le D \le 800$	0.04 – 0.045

- FROM THE CURVED BEAM THEORY, THE RADIAL DISPLACEMENT (u) AND THE MAXIMUM BENDING STRESS (σ_b) IN THE RING CAN BE COMPUTED AS:

$u = 3\pi \frac{F R_m^3}{EI} = \frac{3\pi}{8} \frac{F D_m^3}{EI}$ (FROM THE CURVED BEAM THEORY)

$\sigma_b = \frac{F D_m}{W_b} = \frac{F D_m}{I/(a/2)} = \frac{F D_m a}{2I} = P_{ADM} \Rightarrow I = \frac{F D_m a}{2 P_{ADM}} \Rightarrow u = \frac{3\pi}{8} \frac{F D_m^3}{E \frac{F D_m a}{2 P_{ADM}}} = \frac{3\pi}{4} \frac{D_m^2 P_{ADM}}{E a}$

- BY EXPRESSING THE MOMENT OF INERTIA (I) AS A FUNCTION OF THE MAXIMUM BENDING STRESS SET EQUAL TO THE ADMISSIBLE PRESSURE, AND SUBSTITUTING IN THE RADIAL DISPLACEMENT EXPRESSION, IT BECOMES:

$u = \frac{3\pi}{4} \frac{D_m^2 P_{ADM}}{E a}$ (MAXIMUM RADIAL DISPLACEMENT)

- THE TOTAL FREE GAP (m) CAN BE COMPUTED AS: APPLYING THE RING THEORY THAT IS A MODIFICATION OF THE CURVED BEAM THEORY $\rightarrow m$

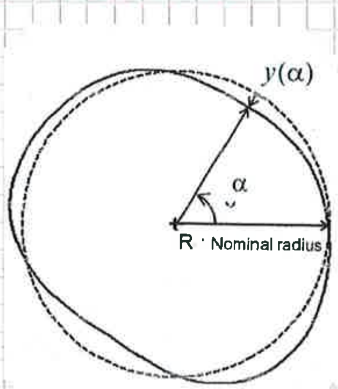
$m = \frac{9\pi}{4} \frac{P_{ADM} D_m \left(\frac{D_m}{a} - 1\right)^3}{E}$ (FROM RING THEORY, TO VERIFY FIRST ATTEMPT m)

- THE MAXIMUM STRESS DUE TO THE REDUCTION OF THE RING DIAMETER TO THE WORKING VALUE (BORE) IS:

$\sigma' = \frac{E}{\frac{3\pi}{4}\left(\frac{D_m}{a} - 1\right)} \frac{m}{a}$ (MAX STRESS IN COMPR. RING CONF.) (FROM THE RING THEORY)

- IT CAN BE ASSUMED THAT DURING THE ENGINE WORKING CONDITION THE PISTON RING DEFORMATION IS SIMILAR TO THE LINER DISTORTION; THEREFORE A SUFICIENT ACCURACY WHEN INTERPOLATING THE LINER DISTORSION SHOULD MEAN A SUFFICIENT ACCURACY ON THE PISTON RING DEFORMATION.
- A CURVE REPRESENTATIVE OF THE DEFORMED TRASVERSAL CROSS SECTION OF THE LINER IS THE HERMITE 5th ORDER POLYNOMIAL SPLINE.
- DUE TO THE COMPLEXITY OF THE CONTACT INTERACTIONS BETWEEN PISTON RING, PISTON AND LINER, FE ANALYSIS IS REQUIRED TO PREDICT THE PISTON RING BEHAVIOUR.
- A NEW CURVED BEAM FINITE ELEMENT HAS BEEN PRESENTED IN LITERATURE (BAELDEN C., TIAN T., A DUAL GRID CURVED BEAM FINITE ELEMENT MODEL OF PISTON RINGS FOR IMPROVED CONTACT CAPABILITIES, SAE INT. J. ENGINES, 2014, 7(1): 156-171, DOI: 10.4271/2014-01-1085) AND SPECIFICALLY DEVELOPED TO COUPLE THE PISTON RING STRUCTURAL DEFORMATION WITH THE LINER DISTORSION.
- TO DESCRIBE THE DEFORMED SHAPE OF THE LINER TRANSVERSAL SECTION, THE FOURIER SERIES CAN BE USED:

$$y(\alpha) = A_0 + \sum_{k=1}^{m} A_k \sin(k\alpha + \phi_k)$$



WITH:

α : POSITION ALONG THE LINER CIRCUMF.

A_k AND ϕ_k : MAGNITUDE AND PHASE OF THE K-th DISTORSION ORDER.

- AS 0-th ORDER OF DISTORSION CORRESPONDS TO A UNIFORM EXPANSION OF THE LINER RADIUS AND 1-st ORDER OF DISTORSION CORRESPONDS TO A RIGID DISPLACEMENT OF THE LINER AXIS (WITHOUT MODIFYING ITS RADIUS), 0-th AND 1-st ORDERS OF DISTORSION CAN BE NEGLECTED.
- THE SHAPE OF THE K-th ORDER OF DISTORSION WILL CONTAIN K LOBES EQUALLY DISTRIBUTED AROUND THE LINER CIRCUMFERENCE.

• PISTON RING LUBRICATION

• PISTON RINGS ENJOY DIFFERENT LUBRIFICATION CONDITIONS:
- HYDRODYNAMIC LUBRICATION AT MID-PHASE OF THE ENGINE STROKE
- BOUNDARY LUBRICATION AT CRITICAL POSITION (TDC AND BDC)
- MIXED LUBRICATION NEARBY TO CRITICAL POSITIONS. (MORE COMPLICATED)

⊙ IN HYDRODYNAMIC LUBRICATION (OR FULL FILM LUBRICATION), OIL FILM IS SUFFICIENTLY THICK TO SUSTAIN LOADS AND ASPERITY CONTACT IS NEGLIGIBLE.

⊙ IN BOUNDARY LUBRICATION, OIL FILM IS THIN AND THE LOAD IS SUPPORTED MAINLY OR COMPLETELY BY ASPERITY CONTACTS.

• IN MIXED LUBRICATION (I.E. THE TRANSITION REGION BETWEEN THE TWO MENTIONED LUBRICATION REGIMES), THE LOAD IS SUSTAINED BY BOTH LUBRICANT FILM AND ASPERITY CONTACTS; MIXED LUBRICATION TAKES PLACE WHEN THERE IS HIGH LOAD, LOW SPEED, AND/OR LOW VISCOSITY DUE TO HIGH TEMPERATURE.

• TO SIMPLIFY PISTON RING-PACK TRIBOLOGY PROBLEM, IT IS COMMON TO MAKE ASSUMPTIONS DURING ANALYTICAL MODELING AND NUMERICAL SIMULATION.

hp1 - OIL IS ASSUMED AS NEWTONIAN AND INCOMPRESSIBLE FLUID.

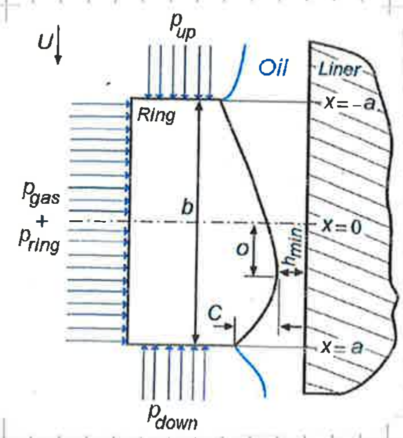
hp2 - OIL IS TREATED AS AN ISOVISCOUS FLUID.

hp3 - CIRCUMFERENTIAL VARIATIONS IN OIL FILM THICKNESS ARE NEGLECTED (OIL FILM THICKNESS CIRCUMFERENTIAL SYMMETRY IS ASSUMED) BUT OIL FILM THICKNESS CHANGES IN RING MOTION DIRECTION, ENCOURAGING USE OF 1D REYNOLDS EQUATION.

hp4 - OIL FLOW IS LAMINAR

hp5 - THERE IS NO SLIPPAGE AT THE BOUNDARIES

hp6 - RING DOESN'T TILT IN ITS GROOVE DURING OPERATING CONDITION, OR IT MAY ROTATE AT VERY LOW FREQUENCY.



- FORCE BALANCE EQUILIBRIUM EQUATION IN DOWNWARD STROKE:

$$\int_0^a p(x)\,dx + (a+o)P_{UP} = b\left(\frac{2T}{bD} + P_{GAS}\right)$$

- FORCE BALANCE EQUILIBRIUM EQUATION IN UPWARD STROKE:

$$\int_{-a}^{o} p(x)\,dx + (a-o)P_{DOWN} = b\left(\frac{2T}{bD} + P_{GAS}\right)$$

WITH:

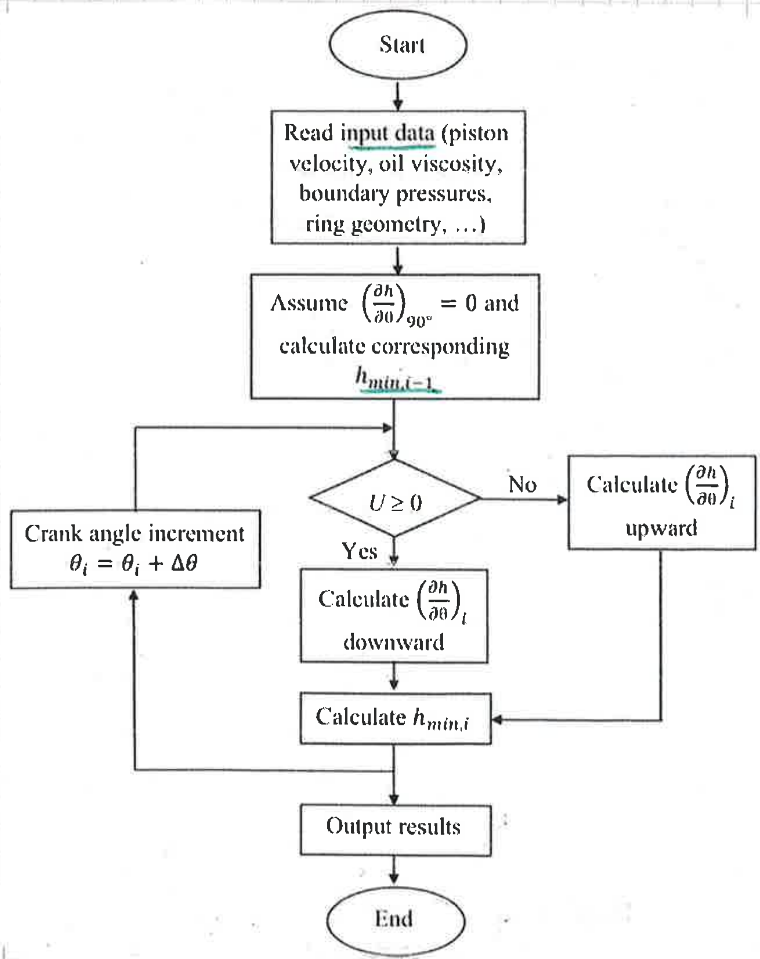
T : RING TENSION FORCE

P_{GAS} : PRESSURE ACTING ON THE REAR OF THE RING ; MAXIMUM OF BOUNDARY PRESSURES ACTING ON RING UPPER AND LOWER EDGES :

$P_{GAS} = P_{UP}$ IF $P_{UP} > P_{DOWN}$

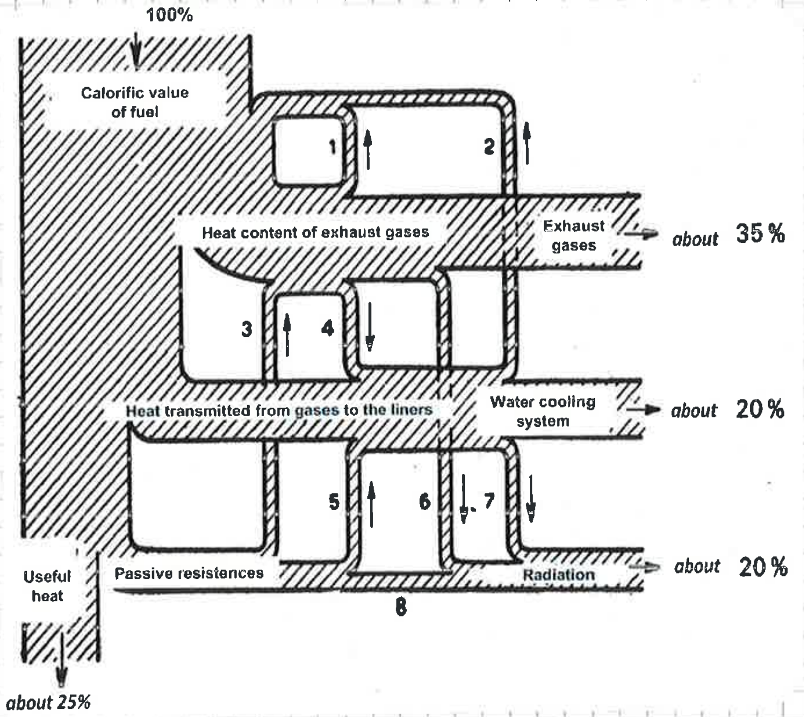
$P_{GAS} = P_{DOWN}$ IF $P_{DOWN} > P_{UP}$

- ALGORITHM FOR SOLVING FORCE EQUILIBRIUM EQUATION TO CALCULATE INSTANTANEOUS MINIMUM OIL FILM THICKNESS (h_{min}) AS A FUNCTION OF CRANK ANGLE (θ).



• NUMERICAL ANALYSIS - PISTON

- BECAUSE **MOST OF THE PISTON MASS** IS **POSITIONED** IN THE **ZONE** WHERE THERE ARE THE **PISTON HUBS**, THE **THERMAL FIELD** ACTING ON THE PISTON **DEFORMS MORE** THIS **HUB ZONE THAN** THE **LOWER PART** OF THE PISTON.
- TO ENSURE A GOOD FITTING BETWEEN THE PISTON AND THE LINER, THE **DESIGN** OF THE LOWER PART OF THE PISTON **MUST TAKE INTO ACCOUNT THIS THERMAL DEFORMATION** OF THE PISTON SKIRT AND ADEGUATE DESIGN STRATEGIES ARE USED (E.G. STEEL INSERTS EMBEDDED IN THE CASTING).
- MOREOVER, **IN DIESEL ENGINES**, THE **IMPORTANT VARIATION IN PRESSURE** DURING THE ENGINE CYCLE CAUSES **FATIGUE LOADS** WHICH BECOME VERY CHALLENGING FOR THE PISTON.
- **JUST ONE PART** (25-30%) OF THE **FUEL THERMAL CAPACITY** IS TRANSFORMED **INTO MECHANICAL POWER** BECAUSE ANOTHER **IMPORTANT AMOUNT** (ABOUT **60%**) **GOES TO STRESS** THE **PISTON** AND THE **LINER**.



4 (H.T. COEFF. VARIATION)

- THE HEAT EXCHANGE MODELLING IS FURTHER COMPLICATED BECAUSE:
  - 4.1 (MOVABLE PISTON) PISTON MOVES ALONG THE LINER AND THE HEAT TRANSFER COEFFICIENTS VARY DURING THE ENGINE CYCLE.
  - 4.2 (MOVABLE RINGS) ELASTIC RINGS MOVE IN THEIR SEATS DURING THE RECIPROCATING MOTION
  - 4.3 (CONV-COND.) THE PRESENCE OF OIL BETWEEN THE ELEMENTS PRODUCES A MIXED CONVECTION-CONDUCTION THERMAL PHENOMENON THAT IS DIFFICULT, IF NOT IMPOSSIBLE, TO MODEL.
  - 4.4 (CONV.>COND. - NOT MEAS. FACTOR) THE OIL SPRAYED INTO THE PISTON AND IN THE COOLING GALLERY MAKES THE CONVECTION DOMINANT WITH RESPECT TO THE CONDUCTION, BUT WITH A NOT MEASURABLE FACTOR.
  - 4.5 (OIL-AIR) IN REALITY THE OIL IS MIXED WITH AIR AND THIS MAKES THE PHENOMENON EVEN MORE COMPLICATED TO DESCRIBE BY USING MATHEMATIC MODELLING.
  - 4.6 (H.T. COEFF. VARY WITH n AND T_{OIL}) FINALLY IT WOULD BE THEN NECESSARY TO KNOW THE HEAT TRANSFER COEFFICIENTS DEPENDING ON THE ENGINE SPEED AND THE OIL TEMPERATURE.

- IN FEA, THE RADIATION PHENOMENON CAN BE NEGLECTED (EVEN IN CASE OF DIESEL ENGINES WHERE THE EMISSIVITY OF THE PARTICLES GENERATED BY THE COMBUSTION PROCESS IS HIGHER THAN IN GASOLINE ENGINES).
- ONLY THE CONVECTION PHENOMENON IS THEN TAKEN INTO ACCOUNT
- BY MEANS OF PRELIMINARY MATHEMATICAL MODELS AND NUMERICAL CFD ANALYSIS, THE TEMPERATURE MAPS AND THE CONVECTION HEAT TRANSFER COEFFICIENTS ARE OBTAINED IN VARIOUS POINTS OF THE PISTON TOP.
- THE CONVECTIVE HEAT TRANSFER COEFFICIENTS THUS OBTAINED ARE THEN CONSIDERED CONSTANTS WITHIN SPECIFIC MACRO-AREAS OF PISTON FE STRUCTURAL MODEL.

CASE STUDY: PISTON OF FIAT 1.6 JTD, 4 CYL.

- CENTRAL BOWL WITH TOROIDAL SHAPE
- 3 ELASTIC RINGS (2 SEALING RINGS, 1 OIL SCRAPER RING)
- COOLING GALLERY POSITIONED AT THE RING-LAND LEVEL
- PISTON-LINER INTERFACE OPTIMIZED WITH ASYMMETRICAL SKIRT PROFILE.

CONSTANT T IN THE BOWL (TARGET) INCREASES THE COMBUSTION PERFORMANCE

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• NUMERICAL ANALYSIS – 3.1 PISTON STATIC THERMAL ANALYSIS

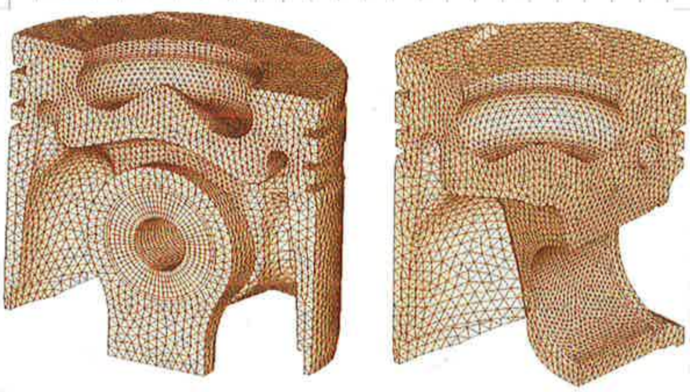
• HALF PISTON IS MODELLED BY SEMI-AUTOMATIC MESHING PROCEDURE:

Component	FE	Nodes	Elements
piston	Tetra4	17690	77220
	Tetra10	121942	
others (connecting rod, pin)	Tetra4	896	2323
	Tetra10	5006	

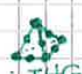
TO TAKE INTO ACCOUNT A LOWER NUMBER OF DOFs WE CAN DIVIDE THE SYSTEM IN SOME MODELS. WE CAN CONSIDER HALF-PIST. IF THERE ARE NOT ASYMMETRIC GEOMETRIES.

• AS IT IS IMPORTANT TO EVALUATE THE STRESSES AND THE THERMAL STATE ALONG THE PIN AXIS AND THE THRUST AXIS OF THE COMBUSTION CHAMBER (ON THE UPPER PART OF THE PIN SEAT AND ON THE CIRCUMFERENTIAL EDGE OF THE BOWL) THE MODEL NODES HAVE TO BE ALIGNED ALONG THE GEOMETRY CONTOURS.

• FOR AN ACCURATE ANALYSIS OF THE THERMO-MECHANICAL LOADS, A NON-UNIFORM ADAPTIVE REFINEMENT PROCESS IS SELECTED: MEAN MESH (2 [mm]), DENSE MESH (0.8 [mm]) FOR TOP, COOLING GALLERY AND PIN SEAT.



WE HAVE TO PUT AT LEAST TWO ELEMENTS TO THE THINNER SIZE OF THE PISTON (AS HERE FOR THE BUSHING)

- TETRA 4, TETRA 10 : 4, 10 ARE ORDER OF SHAPE FUNCTION (= n OF NODES)
- TETRA 4  ⇒ LINEAR SHAPE FUNCTION (BECAUSE WE HAVE JUST 1 NODE FOR EACH VERTEX); WE HAVE NOT THE POSSIBILITY TO DESCRIBE SOME PHENOMENA INSIDE THE ELEMENT (SO THE MATERIAL): TETRAEDRIC ELEMENS SUFFER OF LOCKING PHENOMENON;
- FOR THE THERMAL ANALYSIS, BECAUSE THE EQUATION OF THERMAL EXCHANGE IS A FIRST LEVEL EQUATION (THAT IMPLIES LINEAR SOLUTION) TETRA 4 IS ENOUGH.
- FOR MECHANICAL ANALYSIS TETRA 4 IS NOT ENOUGH, BECAUSE OF THE SECOND LEVEL EQUATION (THAT IMPLIES NOT-LINEAR SOLUTION) → TETRA 10 
- THEREFORE, FIRST WE PERFORM THERMAL ANALYSIS USING TETRA 4; THEN, WE CHANGE TO TETRA 10 FOR STRUCTURAL ANALYSIS.

hp1
- THE STUDY STARTS WITH THE STATIC THERMAL ANALYSIS, EXCLUDING THE THERMAL HEAT TRANSIENT PHASE. (H.T. COEFF. BY LITER. OR CFD AN.)

hp2
- THE INITIAL THERMAL COEFFICIENTS OF THE DIFFERENT PISTON MACRO-AREAS ARE SET FROM LITERATURE DATA, AND ASSUMED CONSTANT OVER TIME NEGLECTING THEIR VARIATIONS DURING THE ENGINE CYCLE. THE ASSUMPTIONS OF STATIC CONDITIONS IMPLIES AN ACCEPTABLE ERROR BECAUSE THE TEMPERATURE OF THE INTERNAL POINTS OF THE PISTON DOES NOT CHANGE SIGNIFICANTLY DURING THE ENGINE CYCLE DUE TO THE THERMAL INERTIA OF THE MATERIAL.
- THE GREATER IS THE NUMBER OF MACRO-AREAS WITH CONSTANT HEAT EXCHANGE PARAMETERS, THE MORE ACCURATE WILL BE THE RESULTS OF THE NUMERICAL SIMULATION.

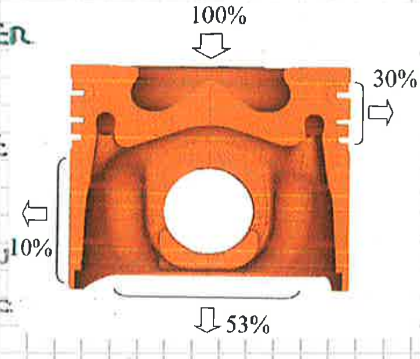
hp3
- IN THE PISTON TOP, THE HEAT TRANSFER IS AFFECTED BY TEMPERATURE AND CONVECTIVE COEFFICIENT WHICH VARIES, ALONG THE PISTON SURFACE, WITH THE MATERIAL CHARACTERISTICS AND THE GAS VELOCITY IN THE BOWL (AND THEREFORE WITH THE ENGINE SPEED).

- IN ORDER TO SIMPLIFY THE CALCULATION, DURING THE STATIC THERMAL ANALYSIS THE CONNECTING ROD AND THE BUSHING ARE NOT MODELLED AND ONLY A VALUE TO REPRESENT THEIR INITIAL TEMPERATURE CONDITION IS ASSIGNED.

1
- THE COEFFICIENTS OF CONDUCTIVE HEAT EXCHANGE AND CONVECTIVE HEAT EXCHANGE ARE CALIBRATED BY USING AN ITERATIVE CALIBRATION PROCEDURE (SEVERAL RUNS UNTIL T-CONVERGENCE)
  - INITIAL DATA FROM LITERATURE ARE CONSIDERED (OR FROM PRELIM. CFD)
  - STATIC THERMAL ANALYSIS IS RUN
  - THE COMPUTED NODAL TEMPERATURES ARE COMPARED WITH EXPERIMENTAL DATA SUPPLIED BY PISTON MANUFACTURERS (OR BY LITERATURE).
  - IN EACH SIMULATION, THE PARAMETERS OF THE THERMAL MODEL ARE CHANGED UNTIL THE MODEL GETS A TEMPERATURE DISTRIBUTION CORRESPONDING TO THE DECLARED ONE.

2
- A FINAL TEST ON THE HEAT FLOW VALUES IS MADE: THE THERMAL POWER ENTERING INTO THE PISTON MUST BE QUITE EQUAL TO THE OUTGOING THERMAL POWER EXCHANGED WITH THE COOLING FLUID AND THE ADJACENT METAL PARTS.

- BY CONSIDERING THE CASE OF THERMAL ISOLATION (IGNORING INTERNAL DEFORMATIONS AND MECHANICAL WORK) THE GLOBAL POWER HEAT ENTERING INTO THE PISTON TOP SHOULD BE EQUAL TO THE SUM OF ALL THE OUTGOING POWER HEAT FROM THE PISTON EXTERNAL SOURCES (WHICH EXCHANGE HEAT WITH FLUIDS AND SURROUNDING MATERIALS).
- COMPARED TO LITERATURE AVAILABLE DATA, THE POWER DISSIPATED IN THE INNER PART OF THE PISTON CAN BE HIGHER THAN THE EXPECTED, TO DETRIMENT OF THE VALUE OF DISSIPATION IN THE SKIRT AND THE RINGS.



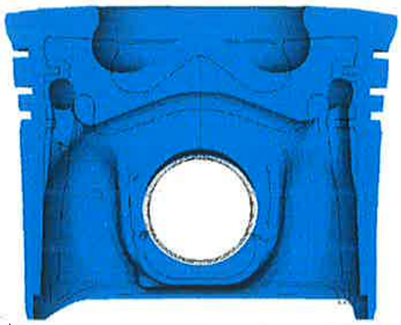
- THIS MAY BE DUE TO ERRORS IN THE T-DISTRIBUTION COMPUTATION AROUND THE WRIST PIN BECAUSE OF DISCRETIZATION ERRORS ON FE SIZE.
- AN ERROR OF ABOUT 7% ON THE HEAT BALANCE (100% IN, 93% OUT) IS CONSIDERED ACCEPTABLE.

- A SENSITIVE ANALYSIS TO THE FE SIZE WAS THAN PERFORMED TO EVALUATE THE AMOUNT OF THE ERROR IN THE HEAT FLUX ESTIMATION.

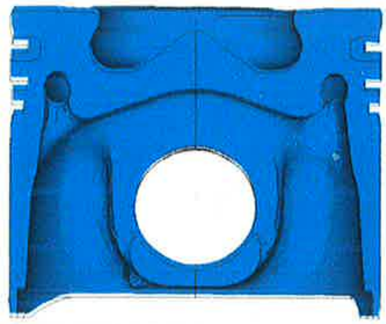
	Error on the heat flux		
size (mm)	Tetra I ord.	Hexa I ord.	Hexa II ord.
7	/	12 %	4 %
5	/	8 %	/
2	3 %	3 %	1 %

- THE RESULTS IN TERMS OF THERMAL AND MECHANICAL DEFORMATIONS PROVIDE THE GUIDE FOR THE CHOICE OF THE DIMENSIONAL TOLERANCE OF THE PISTON DESIGN PHASE.
- THE DIMENSIONAL TOLERANCES SO DEFINED ARE ADEGUATE TO RECOVER THE PISTON DEFORMATION DURING THE OPERATING CONDITION OF THE ENGINE.

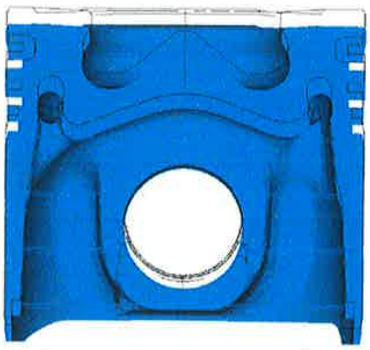
Deformation due to the thermal field



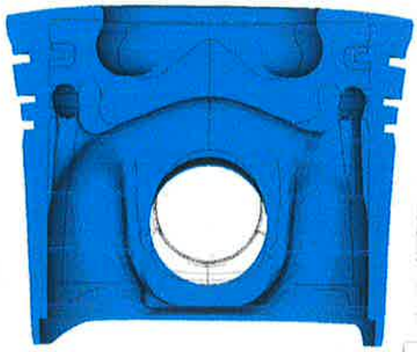
Deformation due to the inertial forces



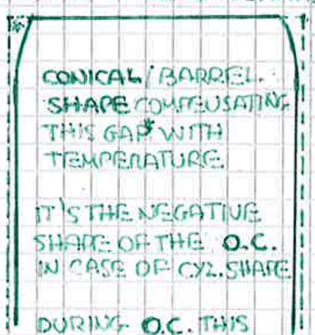
Deformation due to the gas forces



Total deformation



THEREFORE, TO COUNTERACT THE TOTAL DEFORMATION TO AVOID RISK OF PISTON SEIZURE DURING OPERATING CONDITION WE DESIGN A CONICAL SHAPE



DURING O.C. THIS GAP IS RECOVERED BY THE THERMAL EXP.

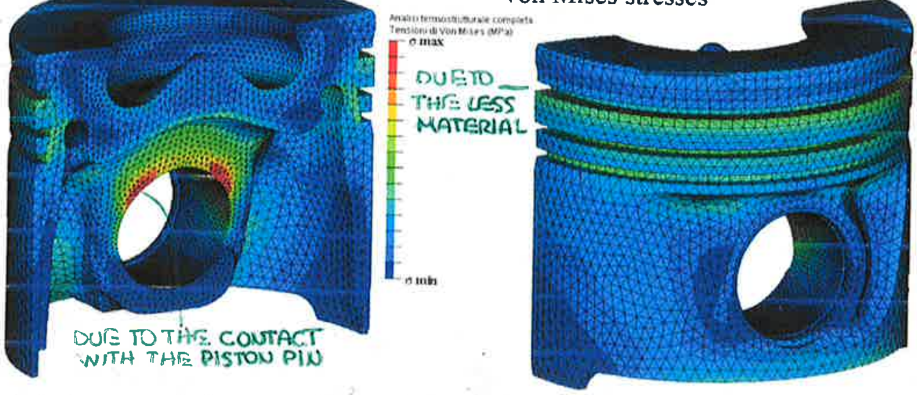
NB: DURING STARTING CONDITION, NOT HAVING THE RIGHT DEFORMATION, WE HAVE PROBLEMS TO CONTROL THE BLOW-BY

THIS CAN BE LIMITATED BY A CORRECT PRELOAD OF THE PISTON RINGS.
SO:
ITS IMPORTANT TO BALANCE THE PISTON DEF. AND THE PRELOAD FORCE KEEPING IN ACCOUNT THAT HIGHER PRELOAD GENERATES HIGHER FRICTION

- THERMO-STRUCTURAL RESULTS : STRAINS AND STRESSES

→ HIGHER STRESSES IN THE PISTON HUB AND RING PACK

von Mises stresses



- THESE RESULTS ARE THE INPUT DATA FOR THE COMPONENT TMF ANALYSIS, DEVELOPED BY USING DIFFERENT DAMAGE MODELS IMPLEMENTED IN SPECIFIC POST-PROCESSING CODES. IF WE OBTAIN HIGHER STRESSES ON THE PISTON TOP AND INSIDE THE MATERIAL IT MEANS THAT WE HAVE NOT A PERFECT COMB. (DETON.) AND CORRECT COOLING

• PISTON SLAP – SIMPLIFIED STUDY WITH FREE BODY DIAGRAMS

• THE SIMPLIFIED STUDY OF THE PISTON SLAP IS BASED ON THE SO-CALLED FREE-BODY DIAGRAMS (THAT CAN BE "SIMPLIFIED" OR "FULL" TYPE) THAT ALLOW:
- TO EVALUATE FORCES AND MOMENTS ACTING ON THE PISTON IN THE DIFFERENT POSITIONS OF THE IMPACT.
- TO WRITE THE EQUATIONS OF MOTION DESCRIBING THE PISTON SLAP.

• IN CASE OF SIMPLIFIED FREE-BODY DIAGRAM, THE FOLLOWING ASSUMPTIONS ARE INTRODUCED:

hp1 – THE ROTATION OF THE PISTON WITHIN THE LINER IS NOT CONSIDERED (NO TILTING, ONLY A PURE TRANSLATION)

hp2 – THE FRICTION FORCES GENERATED IN THE CONTACT PISTON-LINER AND PISTON-PIN ARE NEGLECTED.

hp3 – THE INERTIAS OF PISTON AND PIN ARE NEGLECTED.

hp4 – THE DEFORMATION OF PISTON AND LINER ARE NEGLECTED (BOTH ASSUMED AS RIGID BODIES).

hp5 – THE PRESENCE OF THE OIL FILM IS NEGLECTED.

• TO CALCULATE THE ENERGY DISSIPATED DURING THE IMPACT PISTON-LINER, THE SIMPLIFIED FREE-BODY DIAGRAM IS NOT SUFFICIENT ANT IT IS NECESSARY A LESS APPROXIMATED MODEL.

• IN THE FULL FREE-BODY DIAGRAM, WHILE CONTINUING TO NEGLECT THE DEFORMATION OF THE PISTON AND THE LINER AND THE PRESENCE OF THE OIL FILM, THE FOLLOWING PARAMETERS ARE TAKEN INTO ACCOUNT:
- ROTATION OF THE PISTON WITHIN THE LINER (TILTING).
- FRICTION FORCES PISTON-LINER AND PISTON-PIN.
- PISTON AND PIN INERTIAS.

IN THE FULL FREE-BODY DIAGRAM hp1, hp2, hp3 ARE RELAXED.

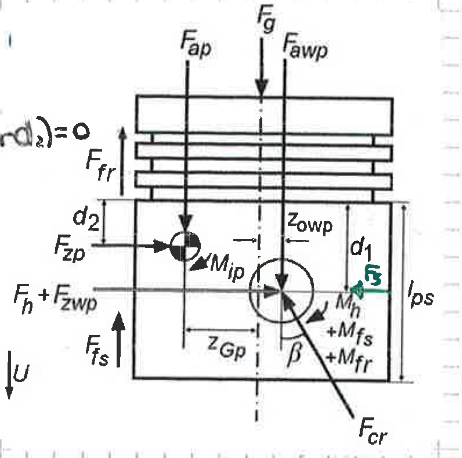
IN NUMERICAL FEA-MBA ALSO hp4, hp5 ARE RELAXED.

- FORCES AND MOMENTS EQUILIBRIUM WITH RESPECT WRIST PIN AXIS. (FULL FBD)

$F_g + F_{ap} + F_{awp} - F_{fs} - F_{fR} - F_{CR}\cos\beta = 0 \rightarrow F_{CR}$

$F_h + F_{zwp} + F_{zp} - F_{CR}\sin\beta = 0$

$M_h + M_{fs} + M_{fR} + M_{ip} - F_{ap}(z_{Gp} + z_{owp}) - F_g z_{owp} + F_{zp}(d_1 - d_2) = 0$



- CALCULATING CONNECTING ROD FORCE (F_{CR}) FROM FIRST EQUATION AND SUBSTITUTING IN THE SECOND ONE:

$F_h - F_s = -(F_{zwp} + F_{zp})$

$M_h - M_s = -(M_{ip} + F_{zp}(d_1 - d_2))$

WHERE:

$F_s = (F_g + F_{ap} + F_{awp} - F_{fs} - F_{fR})\tan\beta$

$M_s = -M_{fs} - M_{fR} + F_{ap}(z_{GP} + z_{owp}) + F_g z_{owp}$

(F_{fs} : SKIRT FRICTION FORCE ; F_{fR} : RINGS FRICTION FORCE ; F_s : THRUST ON THE SKIRT)

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- THE FINAL FIRST-ORDER DIFFERENTIAL EQUATION SYSTEM HAS TO BE SOLVED AT EACH TIME STEP (CRANK ANGLE) APPLYING NUMERICAL METHODS AND ITERATIVE APPROACHES.

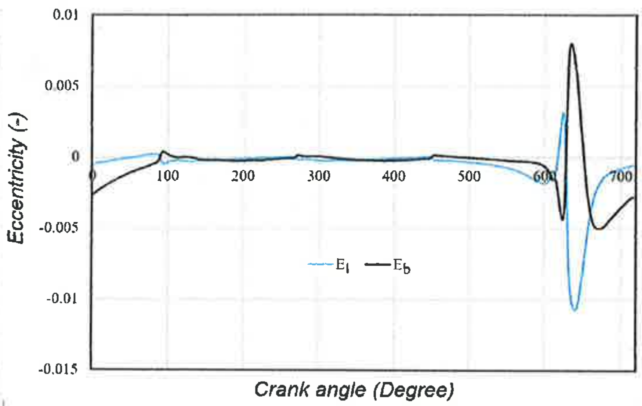
$$\begin{bmatrix} F_h(\dot{e}_t, \dot{e}_b, e_t, e_b) \\ M_h(\dot{e}_t, \dot{e}_b, e_t, e_b) \end{bmatrix} = \begin{bmatrix} F_s \\ M_s \end{bmatrix}$$

- NEWTON-RAPHSON METHOD, THANKS TO ITS CONVERGENCE, ALLOWS THE EVALUATION OF PISTON SKIRT ECCENTRICITIES.

$$\begin{bmatrix} \dot{e}_t \\ \dot{e}_b \end{bmatrix}^{\theta+\Delta\theta} = \begin{bmatrix} \dot{e}_t \\ \dot{e}_b \end{bmatrix}^{\theta} - \begin{bmatrix} \frac{\partial F_h}{\partial \dot{e}_t} & \frac{\partial F_h}{\partial \dot{e}_b} \\ \frac{\partial M_h}{\partial \dot{e}_t} & \frac{\partial M_h}{\partial \dot{e}_b} \end{bmatrix} \begin{bmatrix} F_h - F_s \\ M_h - M_b \end{bmatrix}$$

$$e_t^{t+\Delta t} = e_t^t + \Delta t\, \dot{e}_t$$

$$e_b^{t+\Delta t} = e_b^t + \Delta t\, \dot{e}_b$$



- AFTER EVALUATING PISTON SKIRT ECCENTRICITIES, THE OIL FILM THICKNESS CAN BE UPDATED AT EACH CRANK ANGLE AND USED TO COMPUTE THE HYDRODYNAMIC PRESSURE AT PISTON SKIRT-LINER INTERFACE BY APPLYING THE FINITE DIFFERENCE METHOD (AS FOR BEARING LUBRICATION SOLUTION)

$$\frac{h^3_{i+\frac{1}{2},j}\frac{P_{i+1,j}-P_{i,j}}{\Delta x} - h^3_{i-\frac{1}{2},j}\frac{P_{i,j}-P_{i-1,j}}{\Delta x}}{\Delta x} + \frac{1}{R^2}\,\frac{h^3_{i,j+\frac{1}{2}}\frac{P_{i,j+1}-P_{i,j}}{\Delta\alpha} - h^3_{i,j-\frac{1}{2}}\frac{P_{i,j}-P_{i,j-1}}{\Delta\alpha}}{\Delta\alpha} =$$

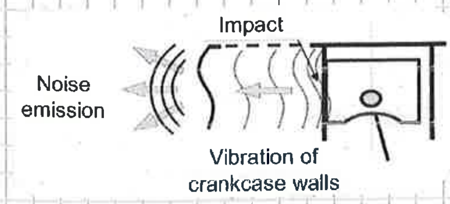
$$= 6\eta U \frac{h_{i+\frac{1}{2},j} - h_{i-\frac{1}{2},j}}{\Delta x} + 12\eta\omega \frac{h^{\theta+\Delta\theta}_{i,j} - h^{\theta}_{i,j}}{\Delta\theta}$$

THEREFORE, KNOWING THE EVOLUTIONS OF THE ECCENTICITIES, WE CAN COMPUTE THE HYDRO-DYNAMIC PRESSURE AND SO THE FORCES

$\sim e_t(\theta), e_b(\theta) \rightarrow h(\theta) \rightarrow F_h(\theta), F_{ss}(\theta), M_h(\theta), M_{ss}(\theta) \rightarrow F_s(\theta), M_s(\theta)$

- PISTON SLAP - EFFECTS

- THE PISTON SLAP EFFECTS ARE RELEVANT IN PARTICUR IN DIESEL ENGINES (HIGH COMBUSTION PRESSURE)



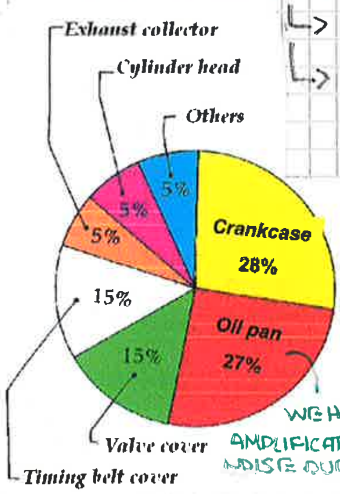
- THE EFFECTS OF THE PISTON SLAP PHENOMENON ARE:

1) EMISSION OF SOUND WAVES FROM THE CRANKCASE SURFACES (THE SO-CALLED "PISTON SLAP NOISE")
2) CAVITATION OF THE COOLANT FLUID
3) SUB-SUPERFICIAL FATIGUE

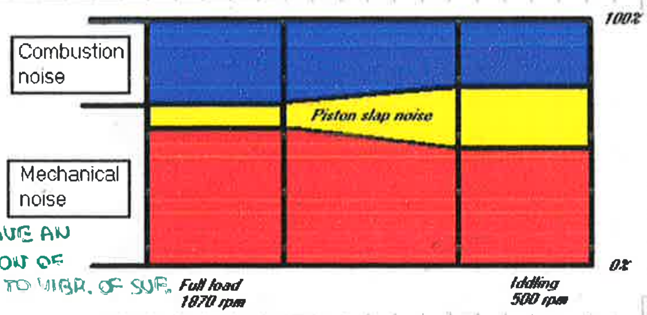
1) PISTON SLAP NOISE

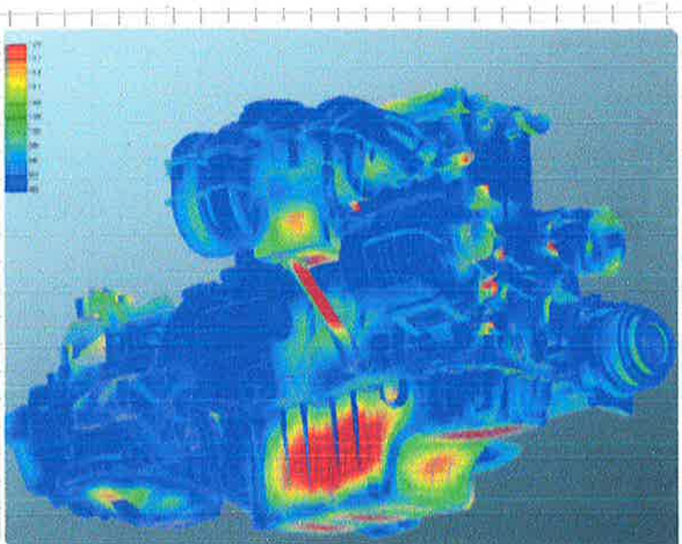
THE CRANKCASE EMITS ABOUT 1/3 OF THE TOTAL SOUND POWER OF ENGINE.

- COMBUSTION NOISE (VIBRATIONS EXCITED BY THE COMBUSTION)
- PISTON SLAP NOISE (IT IS PART OF THE MECHANICAL NOISE)
  ↳ RELEVANT IN IDLE CONDITION AND AT LOW SPIN SPEED.
  ↳ MAXIMUM AT THE ENGINE IGNITION ("IGNITION": AVVIAMENTO)



WE HAVE AN AMPLIFICATION OF NOISE DUE TO VIBR. OF SUF.





MOST IMPORTANT NOISE IS THE MECHANICAL NOISE THEN THE COMBUSTION NOICE FOLLOWED BY THE PISTON SLAP NOISE. (DISTINT FROM MECHANICAL ONE)

WE CAN MEASURE THE PRESSURE COMING FROM THE ELASTO-ACUSTIC WAVE WE CAN EVALUATE THE MOST IMP. ELEMENT WHICH CREATES NOISE → OIL PAN: LARGE SURFACES MOTION

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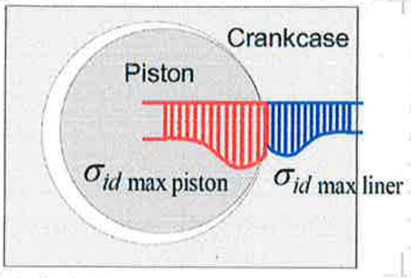
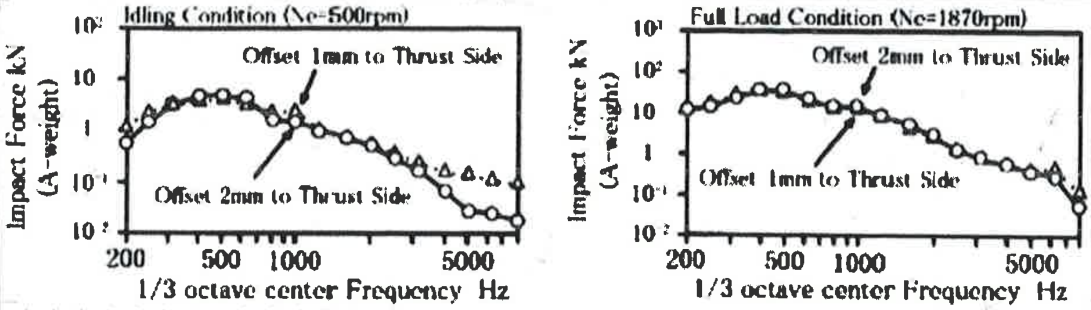
3) SUB-SUPERFICIAL FATIGUE

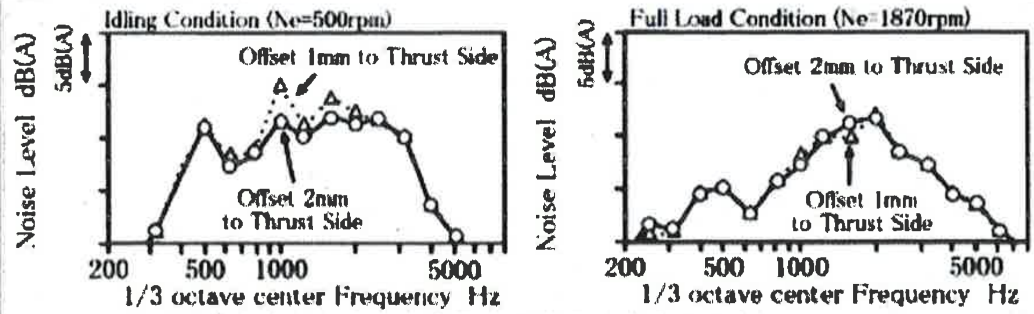
- THE REPEATED IMPACTS BETWEEN PISTON AND LINER PRODUCE HERTZIAN CONTACT FATIGUE.
- SUB-SUPERFICIAL CRACKS OCCUR IN THE LESS RESISTANT COMPONENT AND PROPAGATE UNTIL REACHING THE COMPONENT SURFACE FLAKING OFF MATERIAL PARTICLES (PITTING PHENOMENON).
- IN THE LUBRICANT, THESE METALLIC PARTICLES BECOME DANGEROUS POLLUTANTS AND CREATE SCRATCHES ON THE LINER INTERNAL SURFACE.
- THE POSSIBLE INFILTRATION OF THE COOLANT FLUID INTO THE CRACKS INCREASES THE PROPAGATION SPEED.
- THE PRESENCE OF CASTING INCLUSIONS (METALLIC OR NOT) ALSO AFFECTS THE SUB-SUPERFICIAL STRESS STATE AND PROMOTES THE THE CRACKS INITIATION.



- FROM EXPERIMENTS ON A 6 CYLINDER TURBO-DIESEL ENGINE [NAKASHIMA YAJIMA AND SUZUKI, JSAE REVIEW 20 (1999), pp. 211-216] IT IS CLEAR THAT INCREASING THE OFFSET IN THE DIRECTION OF THE PIN SIDE THRUST INVOLVES:
  - DECREASE IN THE INTENSITY OF THE IMPACT FORCE PISTON-LINER.
  - HIGHER EFFECT IN IDLE CONDITION WITH RESPECT TO FULL LOAD CONDITION
  - EFFECT IS MORE EVIDENT FOR FORCE FREQUENCIES > 5[KHz]



- THE RESULTING DECREASE IN SOUND POWER EMITTED FROM THE CRANKCASE SHOWS:
  - SIGNIFICANT REDUCTION IN IDLE CONDITION FOR LOW SPIN SPEED.
  - STRONG ATTENUATION OF THE PEAKS BETWEEN 1 AND 2 [KHz].
  - NEGLIGIBLE EFFECT AT FULL LOAD CONDITION AND HIGH SPIN SPEED.



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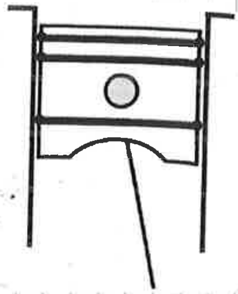
C) ADDITIONAL OIL SCRAPER RING

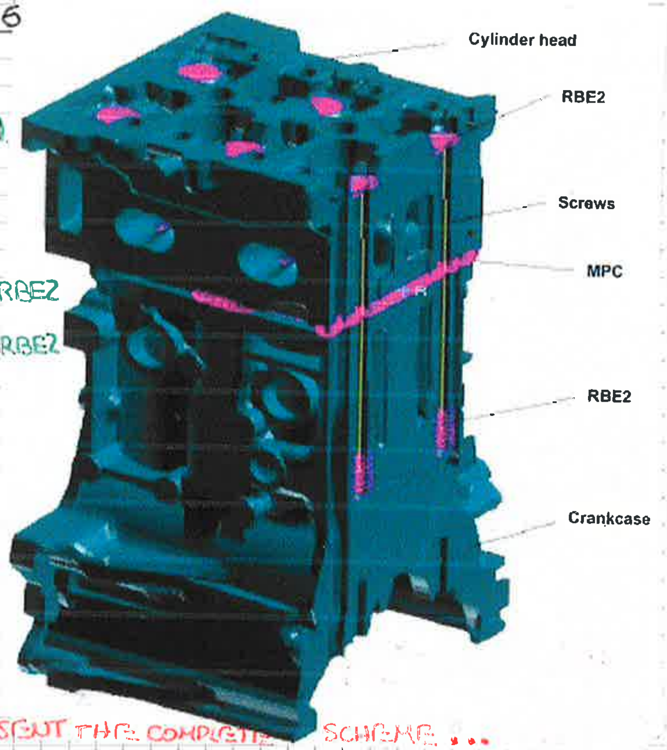


(SMOOTHER W.C.)

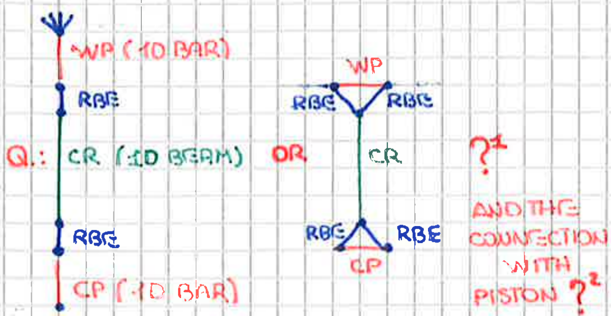
- ADDITIONAL OIL SCRAPER RING IS POSITIONED AT THE BOTTOM OF THE PISTON SKIRT.
- ADVANTAGES:
  - HYDRODYNAMIC LUBRICATION PISTON-LINER IMPROVEMENT
  - DAMPING AND LIFT EFFECTS OF THE OIL FILM.
  - DECREASE OF THE IMPACT FORCE PISTON-LINER.
- DISADVANTAGES:
  - INCREASE OF POWER FRICTION LOSS
  - INCREASE OF PISTON MASS (AND THEN OF THE INERTIAL FORCES).

(FOR TRUCK ENGINE OR INDUSTRIAL ENGINE, WHERE PISTON IS QUITE LONG)

CASE STUDY: FIAT 1.9 JTD, 4 CYL., 16

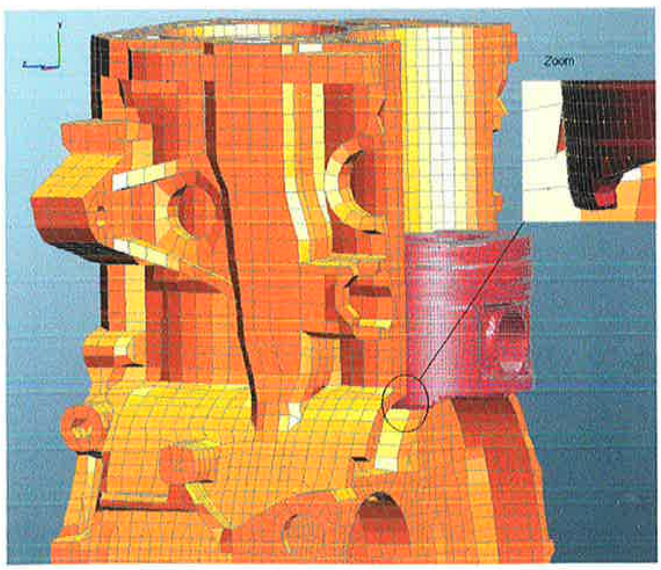


- FE MODEL CONSIDERS:
  - PISTON (FLEXIBLE) (3D MESH: HEXA L. FE)
  - CRANKCASE (FLEXIBLE)
  - CYLINDER HEAD (FLEXIBLE)
  - CONNECTING-ROD (FLEXIBLE) 1D BEAM+RBE2
  - CRANKPIN JOURNAL (RIGID) 1D BAR + RBE2



I HAVEN'T UNDERSTOOD HOW I CAN REPRESENT THE COMPLETE SCHEME...

- THE PISTON IS MODELLED WITH HEXA LINEAR FE AND CONDENSED WITH THE COMPONENTS MODE SYNTHESIS TECHNIQUE. (└ 3D MESH) ... ⇒ CAN YOU DRAW IT ?[3]
- THE DEGREES OF FREEDOM SELECTED AS MASTER DOF ARE:
  - ALL THE NODES OF THE PISTON SKIRT IN CONTACT WITH THE LINER AND THE NODES OF THE PISTON TOP ON WHICH THE GAS FORCE ACTS.
  - ABOUT ONE HUNDRED OF MODAL COORDINATES SELECTED AS ADDITIONAL MASTER DOF IN ORDER TO OBTAIN A GOOD SIMULATION OF THE DYNAMIC BEHAVIOUR OF THE MODEL IN THE HIGH FREQUENCY RANGE.



TO PERFORM THE MBA WE HAVE TO REDUCE DOFs:

$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\}$

IF WE HAVE 1 MILLION DOFs $\{x\}$

→ $[M], [C], [K]$: 1 MILLION × 1 MILLION

⇒ MODE SYNTHESIS TECHNIQUE / CONDENSATION TECHNIQUE, TO REDUCE DOFs.

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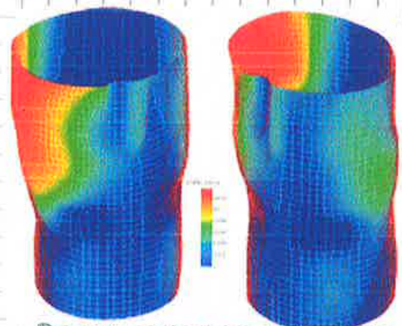
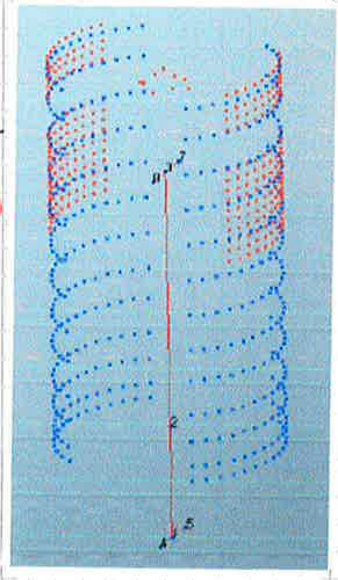
- FOR THE PISTON SLAP NUMERICAL ANALYSIS IT IS NECESSARY TO CREATE AN ACCURATE MODEL OF THE PISTON-LINER COUPLING CONDITION BY USING ELASTO-HYDRODYNAMIC JOINT ELEMENTS.
- IN THE MB MODEL, IN ADDITION TO THE LUBRICANT DATA, AS INPUT DATA ARE ALSO INTRODUCED THE PROFILES (IN COLD CONDITION) OF THE PISTON SKIRT AND THE LINER.
- USING A STATIC ANALYSIS, THE DEFORMED CONFIGURATION OF THE CYLINDER BLOCK IS COMPUTED AND THE OVALIZATION OF THE LINER IS EVALUATED IN A CERTAIN NUMBER OF TRANSVERSAL SECTIONS.
- THE DATA SO OBTAINED ARE THEN INCLUDED IN THE FE CODE WHERE, BY USING AN INTERNAL INTERPOLATION ALGORITHM, THE LINER PROFILE IS COMPUTED IN ALL POINTS OF ITS DEFORMED CONFIG.

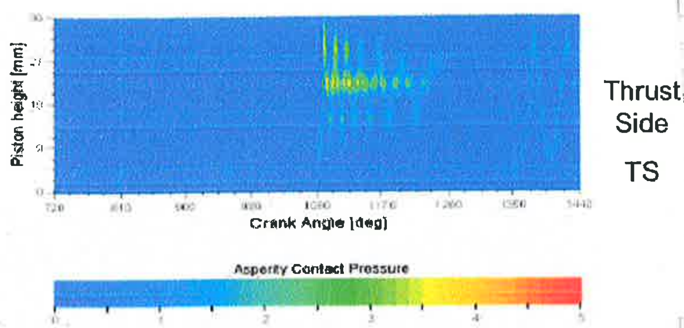

DEFORMATION OF THE LINER.

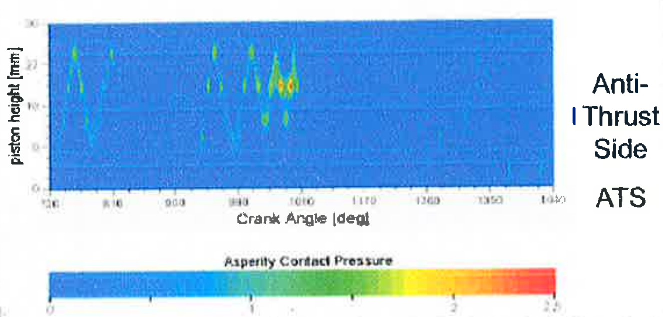
- TO COMPLETE THE MB MODEL THE CRANKPIN IS INSERTED AS A RIGID BODY AND THE MOTION LAW IS IMPOSED IN ORDER TO STUDY THE PISTON SLAP PHENOMENON AT VARIOUS ENGINE SPEEDS.
- THE INPUT LOADS ARE THE TEMPERATURE AND THE PRESSURE TRENDS IN THE BOWL.
- THE ORIGINAL FE MODEL HAS SEVERAL MILLION OF DOFs, WHILE AT THE COND. TECH END, THE CONDENSED MODEL HAS ONLY ONE HUNDRED DOFs.

REDUCED NODES OF THE PISTON
REDUCED NODES OF THE LINER

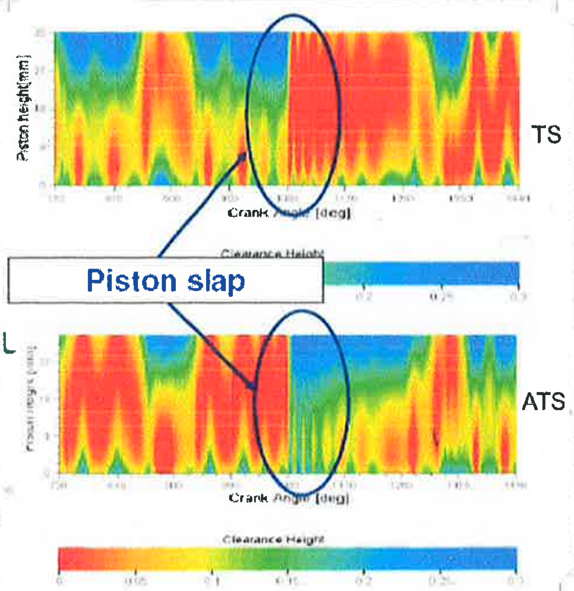


- THE CONTACT PRESSURE PISTON-LINER IS EVALUATED ALONG THE MOST STRESSED PORTION OF THE PISTON SKIRT.
- AN OFF-LINE ANALYSIS OF SUB-SUPERFICIAL FATIGUE CAN BE DONE USING THE COMPUTED CONTACT PRESSURE TRENDS.





- IN OPERATING CONDITION, ON THE MOST STRESSED PORTION OF THE PISTON SKIRT, THE VARIATION OF THE PISTON-LINER GAP IS ALSO EVALUATED. $(h(\theta))$
- FROM THE SIMULATION, IT IS VERIFIED THAT THE MAXIMUM GAP VARIATION IS GREATER THAN THE MAXIMUM GEOMETRICAL GAP BETWEEN PISTON AND THE LINER, THEN THE IMPACT OF THE PISTON ON THE INTERNAL SURFACE OF THE LINER OCCURS. (I.E. PISTON SLAP OCCURS)



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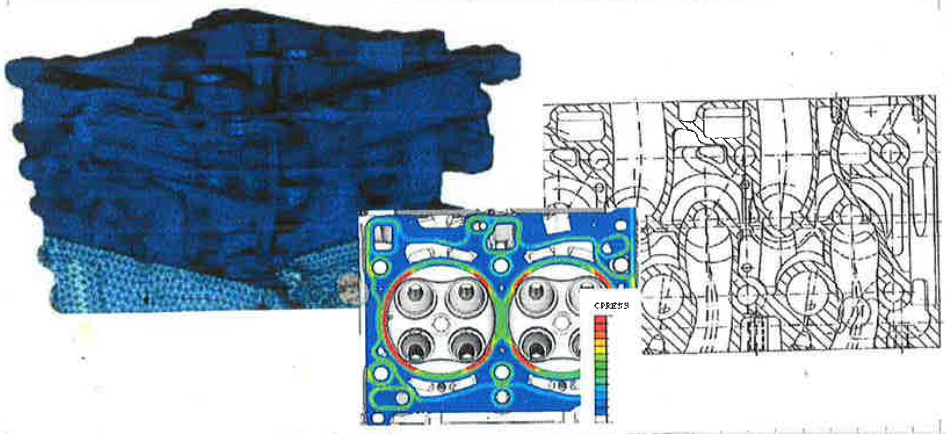
E7 CYLINDER HEAD



- ARCHITECTURE AND GEOMETRY
- COOLING CIRCUIT
- GASKET
- COVER
- RESIDUAL STRESS EXPERIMENTAL INVESTIGATION
- DESIGN GUIDELINES
  - DG - ASSEMBY CONDITION
  - DG - OPERATING CONDITION
- NUMERICAL ANALYSIS (FEA)

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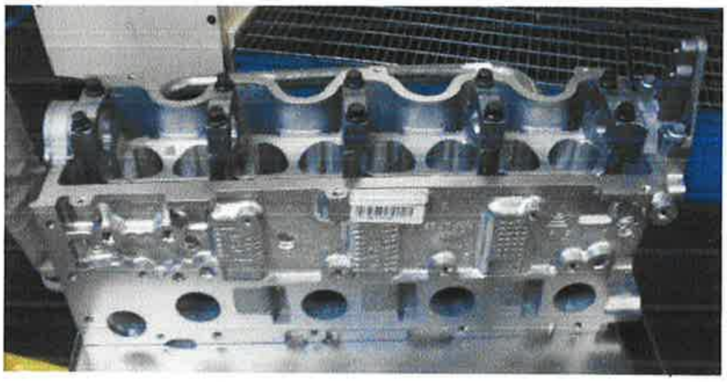
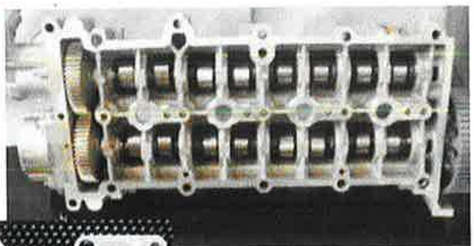
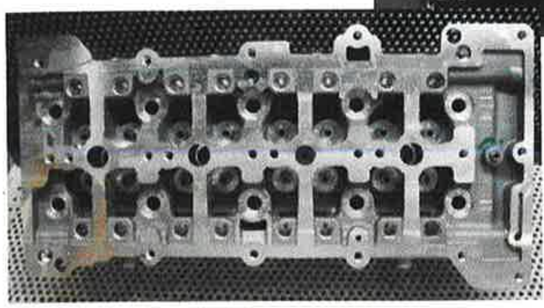
- TWO PARTS ARE EASILY RECOGNIZABLE: A LOWER ONE WHICH CONTAINS:
  - FIRE PLATE FACED TO THE COMBUSTION CHAMBER (BOWL) IN CASE OF GASOLINE IT CONTAINS (ROOF) COMB. CH. FOR EACH CYLINDER
  - INTAKE AND EXHAUST PORTS
  - VALVES HOUSING AND TRACKS
  - COOLING WATER VOLUME AND CIRCUIT.
- AN UPPER PART, WHICH CONTAINS:
  - CAMSHAFT SUPPORTS
  - VOLUME TO HOLD AND DRAIN THE LUBRICANT OIL.
- FOR STIFFNESS REASONS IT IS BETTER THE SOLUTION OF AN INTEGRAL CYLINDER HEAD IN WHICH THESE TWO PARTS ARE OBTAINED IN THE SAME CAST (OBVIOUSLY WITH A MORE COMPLEX GEOMETRY).
- ON THE CONTRARY FOR CONSTRUCTIVE REASONS IT IS BETTER THE SOLUTION WITH TWO SEPARATE PARTS: THE CYLINDER HEAD AND THE CAM-CARRIER. (FOR CASTING REASONS)

- Integral cylinder head (FIAT 1.9 JTD 8v)

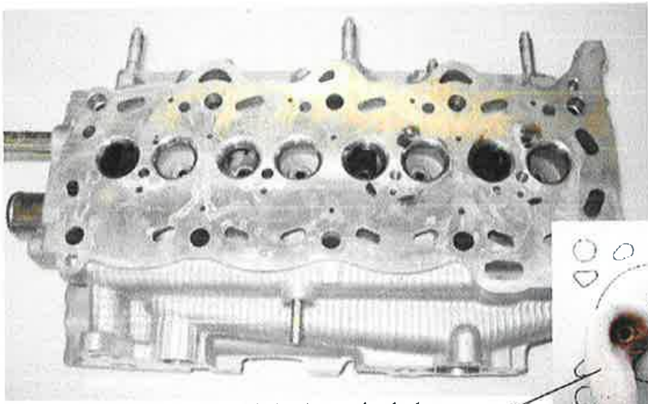


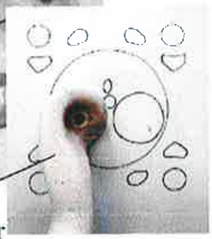
- Cylinder head + Cam-carrier (FIAT 1.9 JTD 16v)





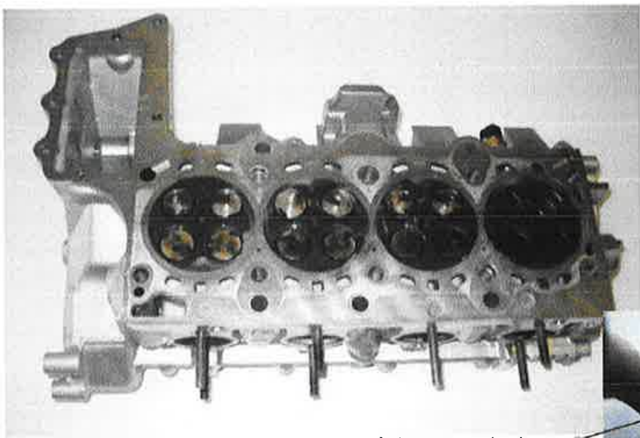
- Toyota Yaris 1.4 D4-D 4cyl 8v





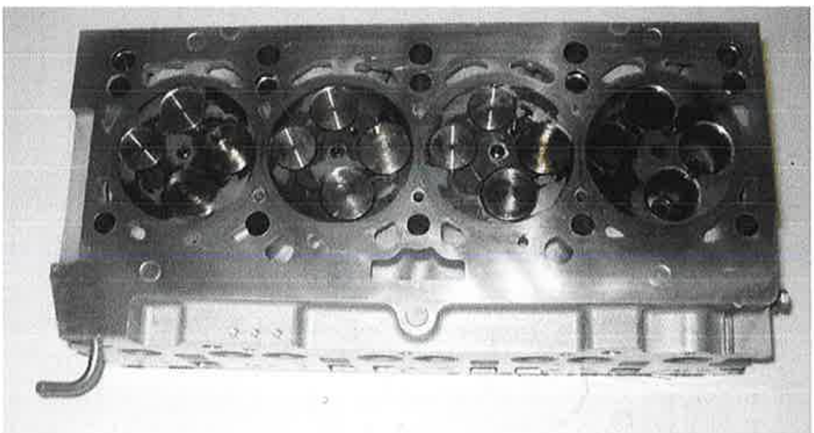
Intake port model: the spiral shape imposes the correct swirl motion to the air

- BMW 120d 4cyl 16v

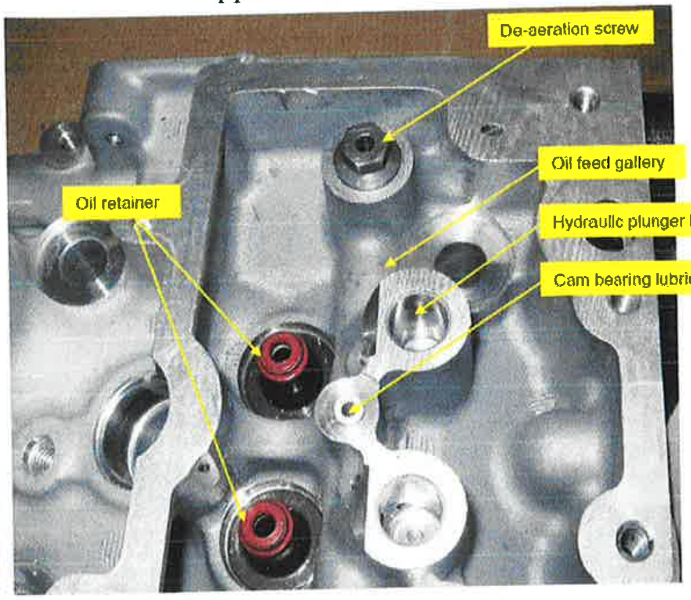


Intake port model: one spiral and one tangential (plunged)

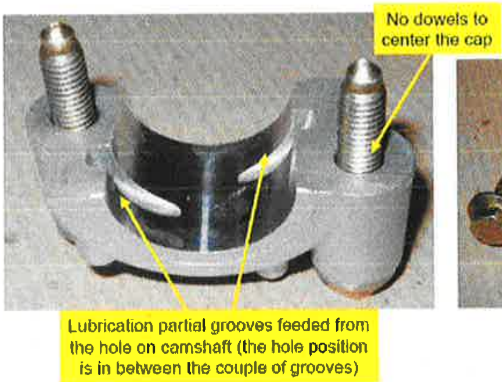
- VW 2.0 TDI 4cyl 16v (cross type valves)



Upper view detail



Caps of camshaft bearing



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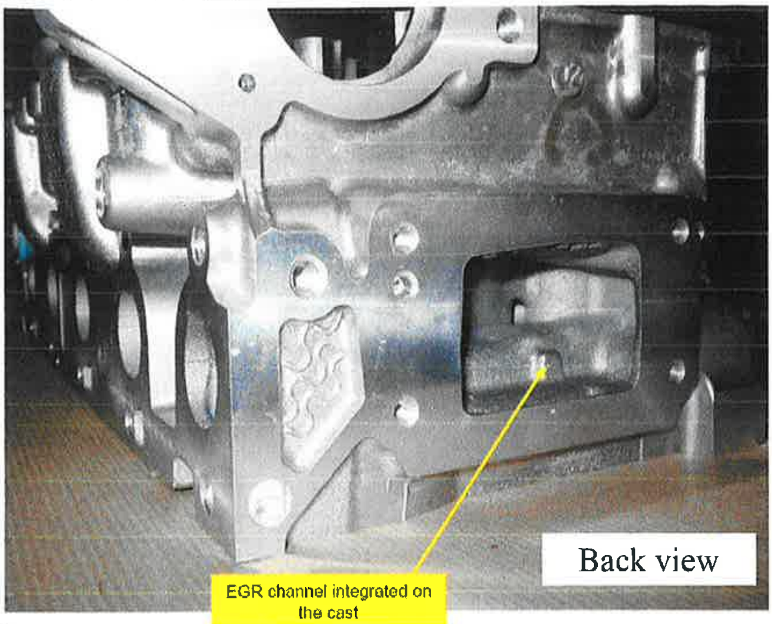
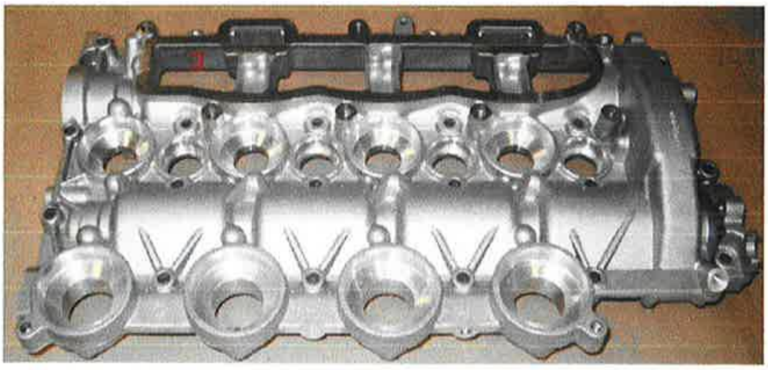
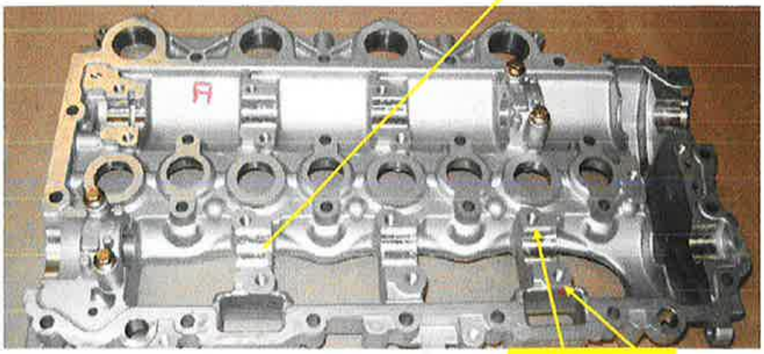


Cam-carrier - Top view



Cam-carrier - Bottom view



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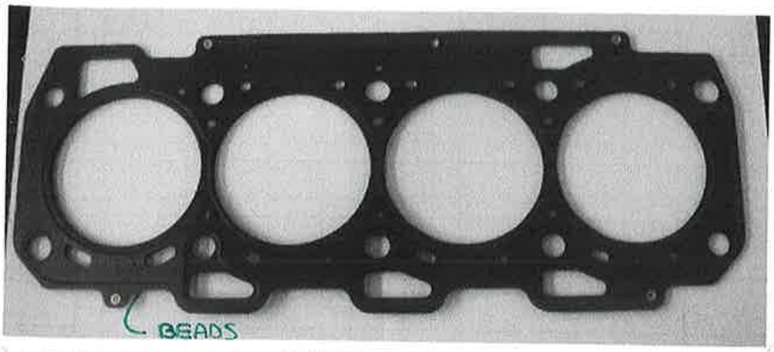
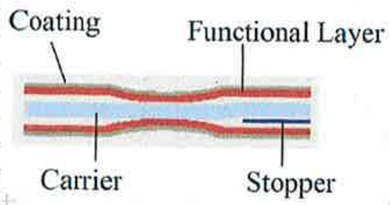
11(3)(4)

- IN THE CROSS-FLOW LAYOUT, ACCORDING WITH THE FRONT DRIVING DIRECTION TWO SOLUTIONS ARE AVAILABLE.
- EXHAUST MANIFOLD POSITIONED ON THE FRONT DRIVING SIDE:
  1. BETTER TEMPERATURES CONTROL ON EXHAUST MANIFOLD AND TURBO-GROUP.
  2. BETTER PROTECTION OF FUEL SYSTEM IN CASE OF FRONTAL BUMP.
  3. POSSIBILITY TO SETUP A FLEXIBLE JOINT SUITABLE TO ISOLATE THE EXHAUST PIPES FROM THE VIBRATION TRANSMITTED BY THE ENGINE.
- INTAKE MANIFOLD POSITIONED ON THE FRONT DRIVING SIDE:
  1. POSSIBILITY TO USE A CLOSE-COUPLED CATALYST WITH LARGE DIMENSIONS
  2. POSSIBILITY TO REDUCE THE LIGHT-OFF TIME OF THE CATALYST (CATALYST IS NOT RUN DOWN ON THE FRONT DIRECTION WIND).
  3. SIMPLIFIED EXHAUST PIPE (IT HAS NOT TO RUN BELOW THE ENGINE)
  4. SIMPLER AND STIFFER OIL PAN, IN CASE OF STRUCTURAL PAN (IT HAS NOT TO CONSIDER THE FLOWING TUNNEL OF THE EXHAUST PIPELINE).

• GASKET





1 • THE HEAD GASKET HAS TO RESIST TO HIGH TEMPERATURE, HIGH PRESSURE AND CORROSION BY GAS, WATER AND LUBRICANT.

2 • MOST COMMON SOLUTION IS A GASKET WITH MULTI-LAYER METAL SHEETS (1-4 SHEETS IN GASOLINE ENGINES, 3-5 SHEETS IN DIESEL ENGINES) WITH AN APPROPRIATE METAL FORMING TO ACHIEVE HIGH STIFFNESS PROPERTY.

3 • FOR DIESEL ENGINES, SPECIAL ATTENTION IS ALSO PAID TO THE CYLINDER BORE SEALING ZONE.

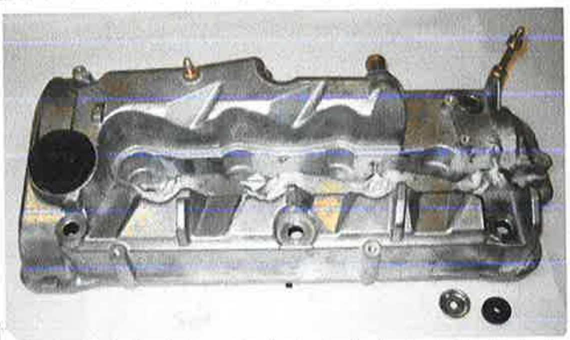
4 • THE GASKET EXTERNAL LAYERS ARE THE FUNCTIONAL LAYERS AND ARE IN CONTACT WITH THE CYLINDER HEAD AND THE CYLINDER BLOCK; THEY ARE MADE BY HARMONIC STEEL AND ON THEIR SURFACE SEVERAL BEADS ARE PRESENT TO PROPERLY DISTRIBUTE THE LOAD ON THE SEALING AREAS. (BEADS <-> RIDGES; AROUND THE BORE)

5 • THE INTERNAL LAYERS (NAMED CARRIERS) ARE MADE BY COMMON STEEL OR LAMINATED STEEL, WITH THE AIM TO:
- GIVE THE ADEGUATE STRUCTURAL STIFFNESS TO THE GASKET.
- DEFINE THE GASKET THICKNESS EQUAL TO THE VALUE REQUIRED BY THE COMPRESSION RATIO OR, EVENTUALLY, TO A DIFFERENT VALUE THAT GUARANTEES THE CORRECT SEALING (EVEN IN CASE OF COUPLED SURFACES SEPARATION DUE TO HIGHER PERFORMANCE ENGINES)
- SUPPORT THE INTERNAL RINGS (NAMED STOPPER)

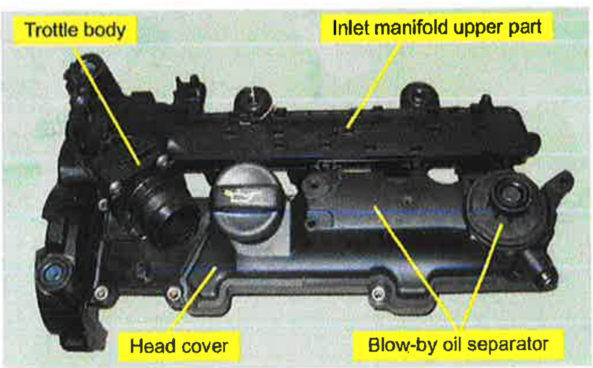
6 • FINALLY AN EXTERNAL RUBBER COATING COVERS THE WHOLE GASKET.

- COVER

- HEAD COVER CAN HAVE THE SIMPLE FUNCTION TO CLOSE THE CYLINDER HEAD OR IT CAN ALSO ACHIEVE SEVERAL OTHER FUNCTIONS AS:
  - SPLITTING VOLUME BETWEEN LUBRICANT AND BLOW-BY GAS.
  - SEALING OF A PART OF THE INTAKE PORTS
  - CAPACITY OF THE AIR FILTER.
- IT IS MADE BY:
  - STEEL - HIGH STIFFNESS, CHEAP, ACOUSTICALLY NOISY
  - Al-ALLOY - HIGH STIFFNESS, LOW WEIGHT, ACOUSTICALLY NOISY
  - Mg-ALLOY - HIGH STIFFNESS, LOW WEIGHT, APHONIC, EXPENSIVE
  - PLASTIC - LOW STIFFNESS, LOW WEIGHT, APHONIC, CHEAP
- TO AVOID THAT THE ENGINE VIBRATIONS ARE TRANSMITTED TO THE CYLINDER HEAD, CAUSING NOISE AS LOUDSPEAKER, THE FASTENING OF THE HEAD COVER ON THE CYLINDER HEAD IS MADE BY SCREWS WITH RUBBER RINGS ON THE STEM.



- Plastic head cover with several integrated functions (PSA 1.4 HDi 8v)



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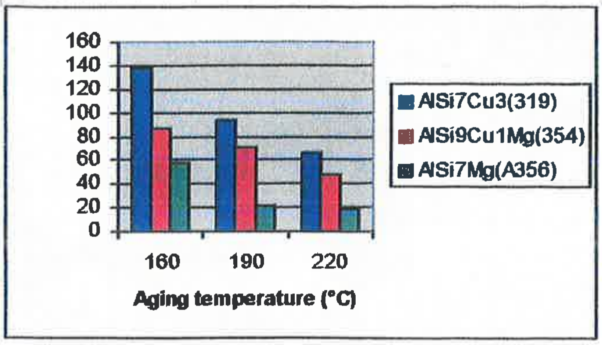
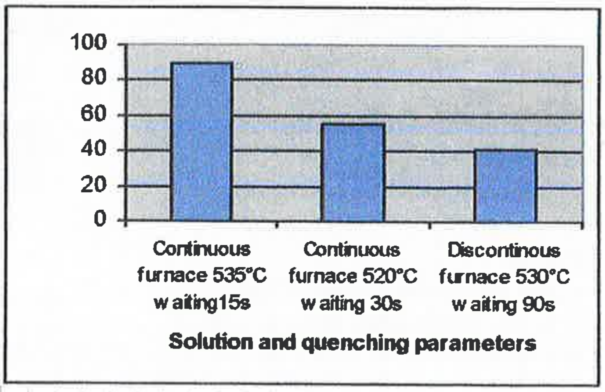
- FOR THE SAME Al-ALLOY, RESIDUAL STRESS DECREASES WITH AGING TEMPERATURE INCREASE. T_{AGING} ↗ : RESIDUAL STRESS ↘

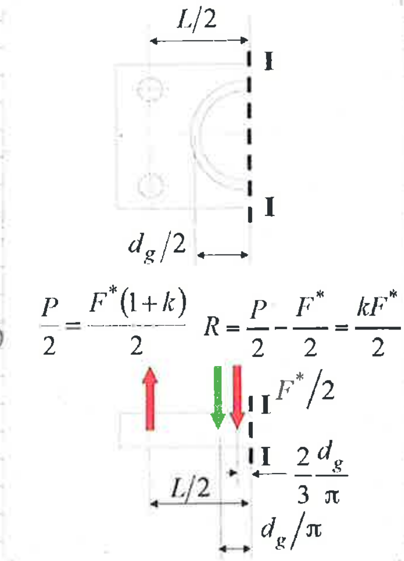


- FOR THE SAME AGING TEMPERATURE, RESIDUAL STRESS MAINTAINS THE FOLLOWING ORDER : AlSi7Cu3 > AlSi9Cu1Mg > AlSi7Mg

- BY INCREASING THE WAITING TIME BETWEEN CASTING AND WATER COOLING, AlSi7Mg CYLINDER HEAD SHOWS AN IMPORTANT REDUCTION OF RESIDUAL STRESS. $t_{WAITING}$ ↗ : RESIDUAL STRESS ↘



- DG - OPERATING CONDITION



- IN OPERATING CONDITION TWO LOADING FORCES ARE PRESENT ON THE CYLINDER HEAD:
  - FIXING PRE-LOAD (RED)
  - LOAD DUE TO THE GASES (RED) WHICH ACTS IN THE CENTER OF MASS OF A SEMI-CIRCULAR AREA AT A DISTANCE $2d_g/(3\pi)$ FROM THE CYLINDER HEAD SECTION I-I.
- OBVIOUSLY THE GASKET REACTS (GREEN) TO BOTH THESE LOADS.
- THE CORRESPONDING BENDING MOMENT IS:

$$M_b = \frac{P}{2}\cdot\frac{L}{2} - R\cdot\frac{d_g}{\pi} - \frac{F^*}{2}\cdot\frac{2d_g}{3\pi} = P_{MAX}\frac{\pi d_g^2}{8}\left[(1+k)\frac{L}{2} - \left(k+\frac{2}{3}\right)\frac{d_g}{\pi}\right]$$

- THE CORRESPONDING BENDING STRESS IS:

$$\sigma_b^I = \frac{M_b}{W_{I-I}}$$

- IN ADDITIONAL TO THIS BENDING STRESS, IN THE CYLINDER HEAD PART IN CONTACT WITH THE EXHAUST GAS EXISTS ALSO A STRESS DUE TO THE TEMPERATURE GRADIENT:

$$\sigma_T = \frac{E\alpha\Delta T}{2(1-\nu)}$$ E: ELASTIC MOD.; α: THERMAL EXPANSION COEFF.; ν: POISSON COEFF.

- THE TOTAL EQUIVALENT STRESS HAS TO BE LIMITED BY THE MATERIAL RESISTANCE:

$$\sigma_{TOT} = \sigma_b^I + \sigma_T \leq \sigma_{ADM}$$

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• NUMERICAL ANALYSIS (FEA)

• THE STRESSES ON THE CYLINDER HEAD (OPERATING CONDITION) CONCERN:

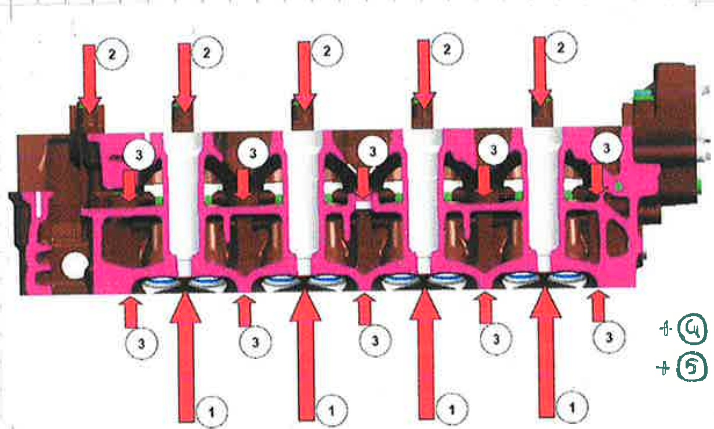
1) LOADS DUE TO THE GAS PRESSURE → FATIGUE, ESPECIALLY ON CAPS LEAN TOWARD THE COOLING FLUID.

2) DYNAMIC LOAD DUE TO THE DISTRIBUTION SYSTEM → FATIGUE, ESPECIALLY ON THE CAMSHAFT SUPPORTS.

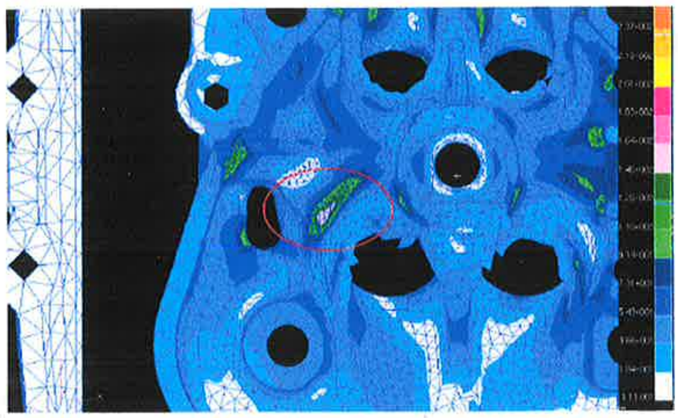
3) TIGHTENING SCREWS TOWARD CYLINDER BLOCK (HOT AND COLD CONDITION) → WORSENING OF PRESSURE TREND ON THE HEAD GASKET.

4) STATIC LOAD DUE TO INTERFERENCE OF ASSEMBLY WITH HOUSING AND VALVES TRACKS → HOUSING VALVES DEFORMATION, ANOMALIES DURING ASSEMBLY, WEAR. (TRACKS = GUIDES)

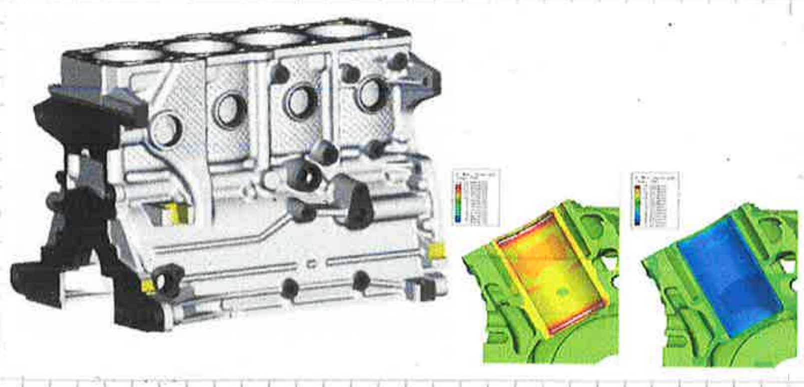
5) THERMAL LOAD (STEADY STATE AND TRANSIENT) → OVERSTRESSES DUE TO HIGH THERMAL GRADIENTS, WORSENING OF THE MECHANICAL PROPERTIES OF LIGHT ALLOYS, WORSENING OF THE COOLING FLUID CIRCULATION.



• STRUCTURAL ANALYSIS: VON MISES STRESSES DUE TO GAS PRESSURE LOAD 1, TIGHTENING FORCE 3 AND THERMAL LOAD 5. WITH A NUMERICAL INVESTIGATION OF THE FE MODEL WE CAN TAKE ALSO INTO ACCOUNT THE DYNAMIC LOAD DUE TO THE TIMING SYSTEM 2 AND THE STATIC LOAD DUE TO INTERFERENCE OF ASSEMBLY 4.

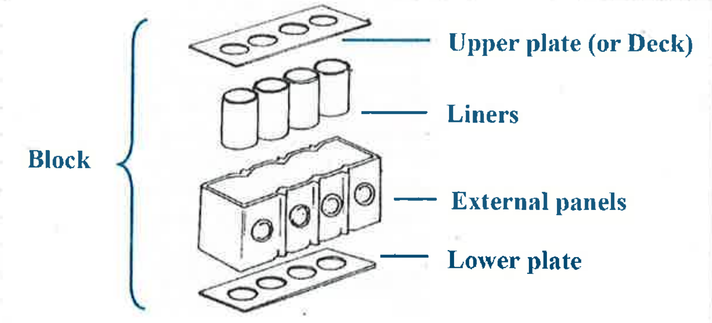


E8 CYLINDER BLOCK



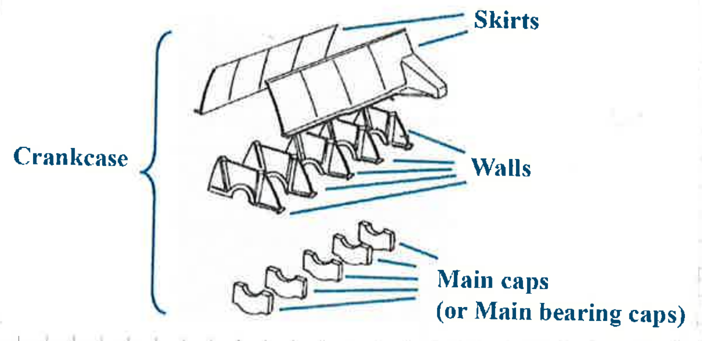
- INTRODUCTION
- ARCHITECTURE AND GEOMETRY
- LINERS
  - L - FINISHING
  - L - COOLING
  - L - MECHANICAL DEFORMATION
  - L - CAST-IN TYPE
  - L - SINTERED TYPE
  - L - BI-LAYER TYPE
  - L - MMC TYPE
  - L - PLASMA COATED TYPE
  - L - CERAMIC COATED TYPE
- FASTENING OF CYLINDER HEAD AND MAIN CAPS
- RESIDUAL STRESS EXPERIMENTAL INVESTIGATION
  - RSEI - RESULTS
- DESIGN GUIDELINES
  - DG - WALL
  - DG - MAIN CAP
  - DG - LINER
  - DG - LINER SUPPORT AND SUPPORT COLLAR
  - DG - LINER WALL
  - DG - LINER FLANGE
- NUMERICAL ANALYSIS (FEA)

- THE BLOCK IS CONSTITUTED BY THE UPPER (DECK) AND THE LOWER PLATES, THE LINERS AND THE EXTERNAL PANELS.

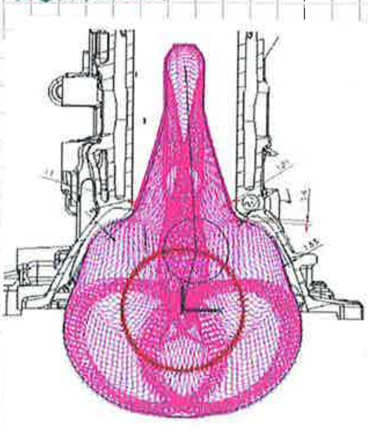


- IT IS LINKED TO THE CRANKCASE AND TO THE CYLINDER HEAD BY MEANS OF SCREWS OR STUDS WHICH GUARANTEE THE CONTROL OF THE MECHANICAL DEFORMATION OF THE LINERS.

- THE CRANKCASE IS CONSTITUTED BY THE SKIRTS, THE WALLS AND THE MAIN CAPS (MAIN BEARING CAPS) WHICH ARE FIXED TO THE WALLS BY MEANS OF SCREWS.



- THE SKIRTS ARE POSITIONED 5-6 [mm] FAR FROM THE "GUITAR" IN ORDER TO ACCOMODATE THE ENVELOPE OF THE CRANK MOVEMENTS (NAMED ALSO "ROD REVOLUTION") AND TO AVOID OIL PUMPING EFFECTS DURING THE CRANK ROTATION.



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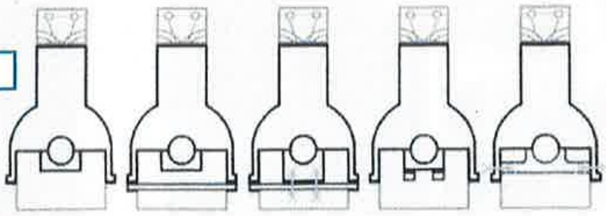
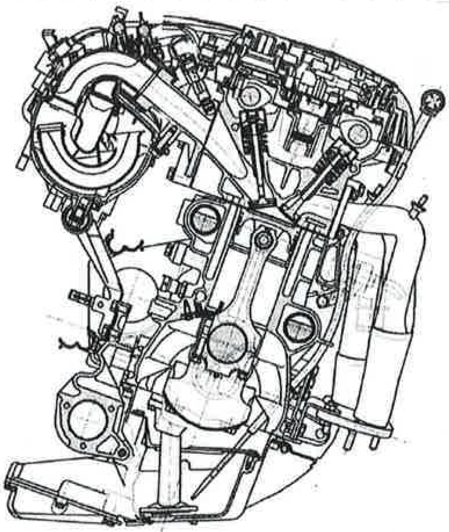
Deep Skirt

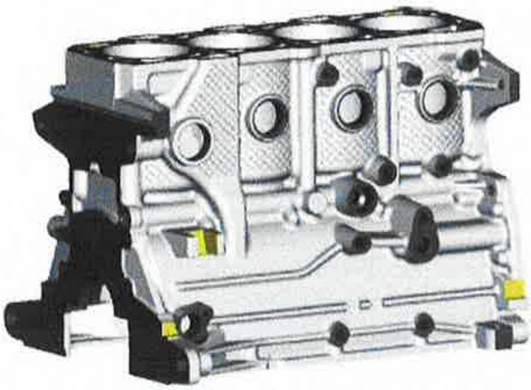


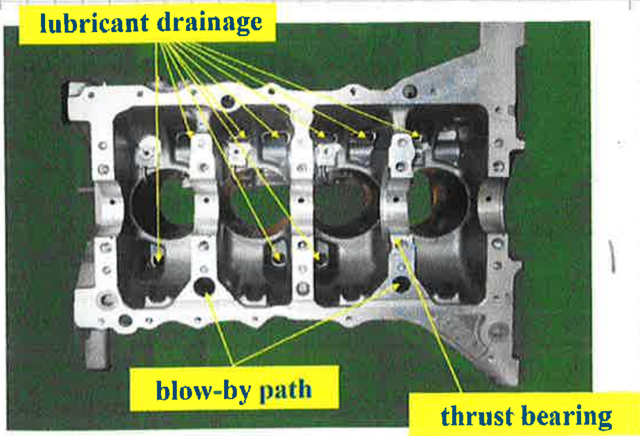
- DEEP SKIRTS EXTEND 60-70 [mm] BELOW THE CENTER LINE OF THE CRANKSHAFT
- PROBLEMS ABOUT NVH (SKIRTS BEHAVE AS LOUDSPEAKERS).
- OIL PAN IS "SHORT" AND THEREFORE CAN BE LESS RESONANT
- THE BEDPLATE SOLUTION MAKES THE STRUCTURE STIFF (AS THE CRANK MECH. CHAMBER IS COMPLETELY CLOSED) HEAVY BUT ALSO EXPENSIVE.

Cast iron cylinder block with deep skirts (Alfa Romeo 1970 16v)



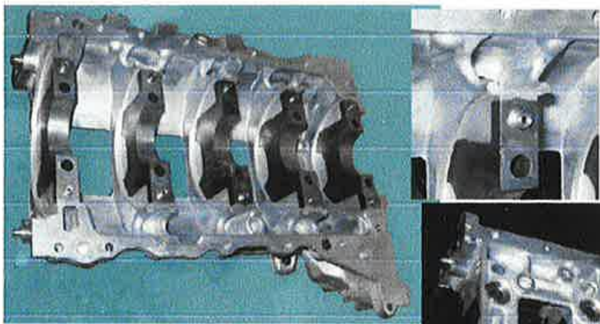
3D model with deep skirts



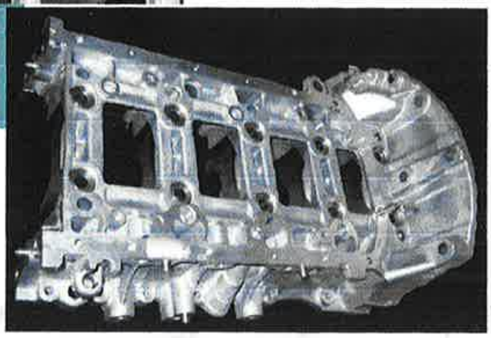


Bottom view

Bedplate with bolted main caps



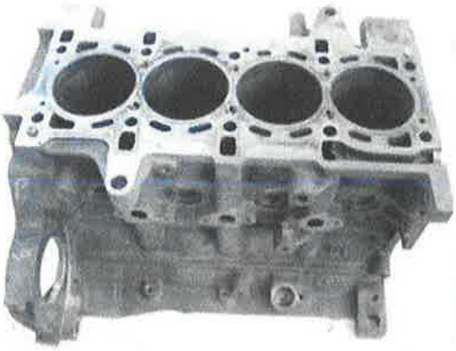
Top view

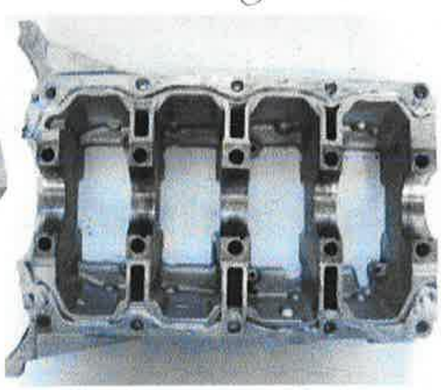


Bottom view

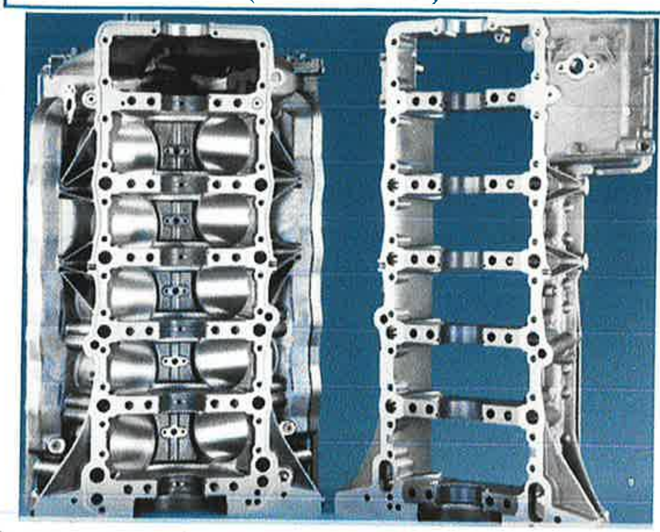
Cast iron cylinder block, Al-alloy bedplate with cast-in main caps (iron) (FIAT SDE 1.3 16v)

3

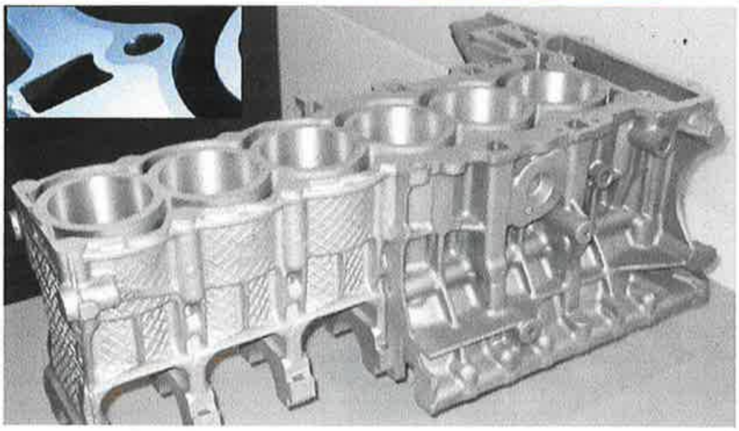




Hypereutectic Al-alloy cylinder block with bedplate
(BMW M5 V10)



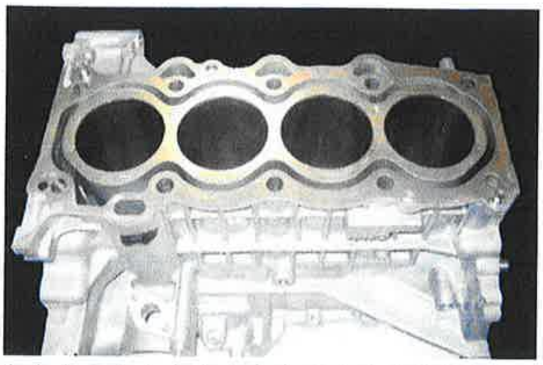
Magnesium cylinder block with Al-alloy inserts
(BMW 3.0 L6 Valvetronic)



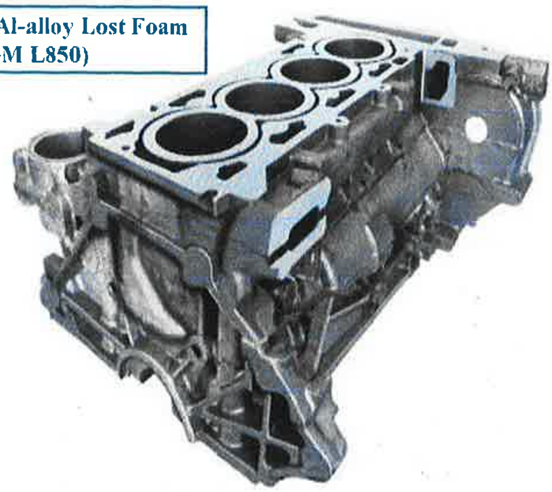
- THE INSERTS ARE CAST FROM HYPEREUTECTIC Al-ALLOY (Si > 12%) IN LOW PRESSURE PERMANENT MOLD AND FORM LINERS, WATER JACKET AND WALLS.
- THE INSERTS ARE REQUIRED TO PREVENT CORROSION PROBLEMS (DUE TO COOLANT FLUID) AND CREEP PROBLEMS OF THE FASTENER JOINTS.
- WITH THIS SOLUTION THE CYLINDER BLOCK IS 45% LIGHTER THEN THE CAST IRON SOLUTION AND 25% LIGHTER THEN THE CONVENTIONAL Al-ALLOY SOLUTION.

- CYLINDER BLOCK WITH CLOSED DECK SOLUTION HAVE AN HIGHER STRUCTURAL STIFFNESS IF COMPARED WITH THE OPEN DECK SOLUTION.
- CLOSED DECK GUARANTEES ALSO A PERFECT SEALING WITH THE CYLINDER HEAD.
- CLOSED DECK SOLUTION PRESENTS A LOWER ACOUSTIC EMISSION COMPARED WITH THE OPEN DECK SOLUTION.

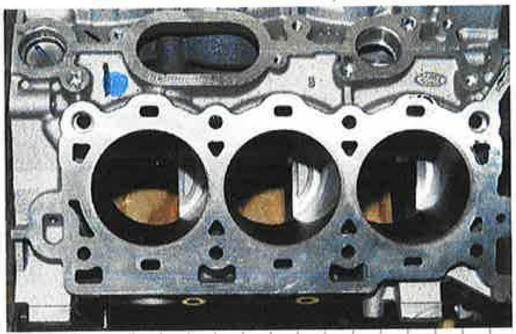
Open deck, Al-alloy HPDC
(Toyota Yaris 1.4 D4-D)



Open deck, Al-alloy Lost Foam
(GM L850)

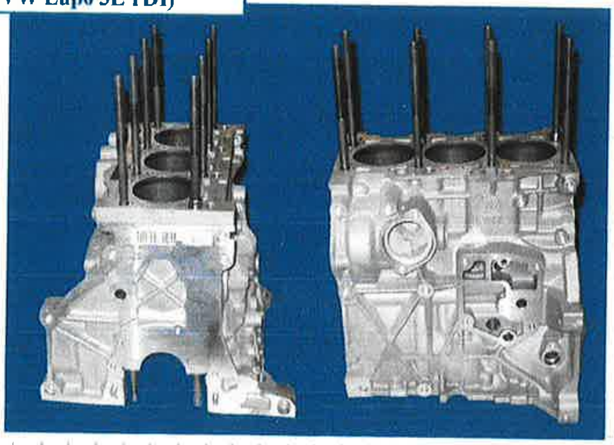


Closed deck
(Ford V6)

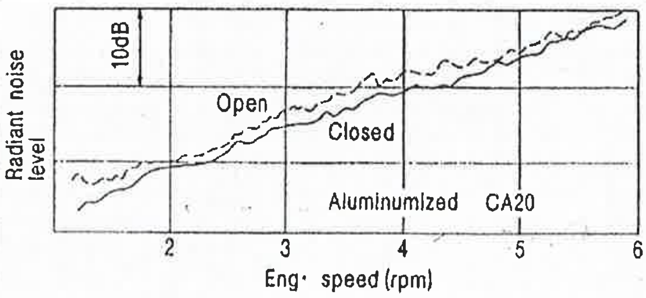


Closed deck, Al-alloy DC
(VW Lupo 3L TDI)

(STUDS)



ACOUSTIC EMISSIONS:



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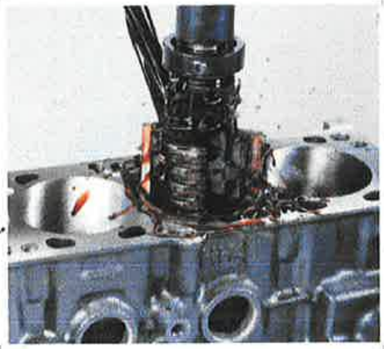
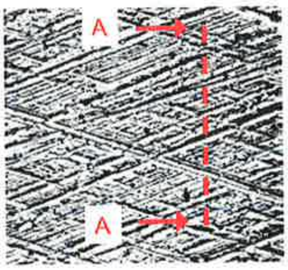
- TO ACHIEVE A GOOD FUNCTIONALITY OF THE LINERS AN **IMPORTANT CONSTRUCTIVE PARAMETER** IS CONSTITUTED BY THEIR **INTERNAL SURFACE FINISHING**:
  - A **GOOD TRIBOLOGICAL COUPLING** BETWEEN LINER INTERNAL SURFACE, PISTON EXTERNAL SKIRT AND PISTON RINGS **LIMITS WEAR** DURING THE RELATIVE MOTION OF THE PISTON.
  - A **GOOD SEALING EFFECT** BETWEEN THESE COMPONENTS, **LIMITS** THE **LUBRICANT CONSUMPTION** AND THE **BLOW-BY PHENOMENON**.
- **ANOTHER IMPORTANT PARAMETER** IS THE **DEFORMATION** OF THE **LONGITUDINAL CYLINDRICAL SURFACE** OF THE **LINER** THAT CAN BE CAUSED BY:
  - **MECHANICAL STRESSES** DUE TO THE **HEAD SCREWS TIGHTENING** AND TO P_g.
  - **THERMAL STRESSES** DUE TO THE **HEAT RELEASED** DURING **COMBUSTION**.

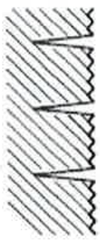
– L – FINISHING

- **PLATEAU FINISHING** USE **SPINDLE** WITH **DIAMOND STONES**
  - **BASIC** PLATEAU FINISHING CREATES **MARKS** (**8-9 [μm] DEPTH**) ACTING AS **POCKETS FOR** THE **LUBRICANT FLUID**.
  - **FINE** PLATEAU FINISHING CREATES **SMALLER MARKS** (**2-3 [μm] DEPTH**) THAT CONSTITUTE THE SUPPORTING SURFACE ON WHICH THE PISTON SLIDES WITH RESPECT THE LINER.



  - IF THE **LINER** IS **TOO POLISHED**, THE **OIL POCKETS** ARE **ABSENT** AND PISTON RINGS SLIDE ON A **TOO DRY SURFACE** AND **TEND TO SEIZE UP**.



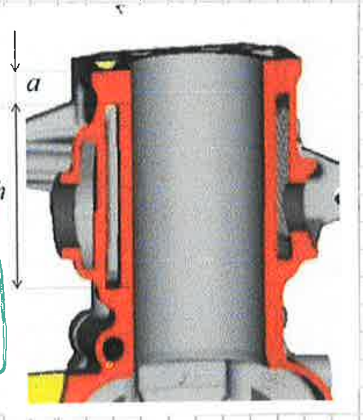


Sez. A-A

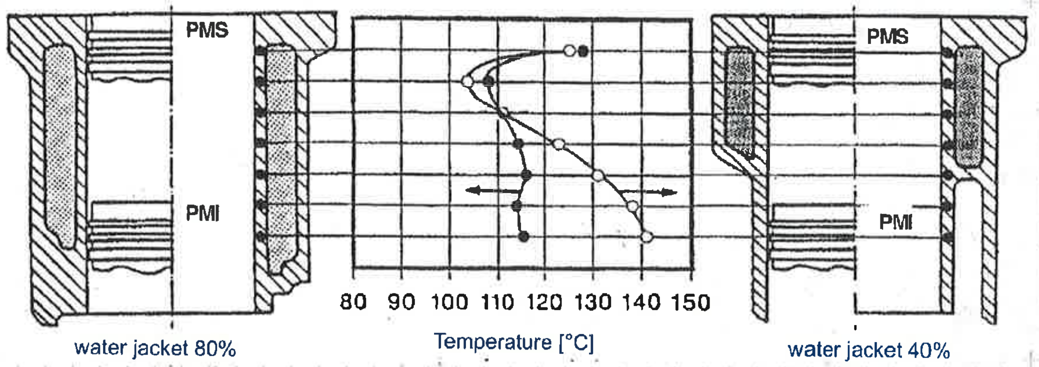
205

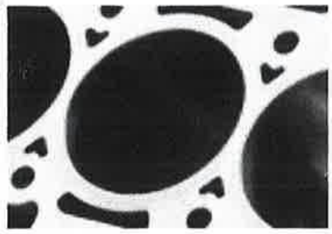
– L – COOLING

- THE LINER COOLING IS GUARANTEED BY THE CIRCULATION OF THE COOLANT FLUID AROUND THE LINERS ALONG THE WATER JACKET THAT IS LOCATED AT THE RING-LAND LEVEL OF THE PISTON.
- TO STIFFEN THE DECK THE WATER JACKET HAS TO BE PLACED IN A LOWER POSITION, INCREASING THE DISTANCE a WITH RESPECT TO THE DECK.
- TO DECREASE THE ENGINE WEIGHT, THE TOTAL LENGTH (h) OF THE WATER JACKET HAS TO BE ALSO REDUCED.
- A NUMERICAL CFD INVESTIGATION IS THEN NEEDED TO AVOID ENGINE WARM-UP PROBLEMS. WHY?

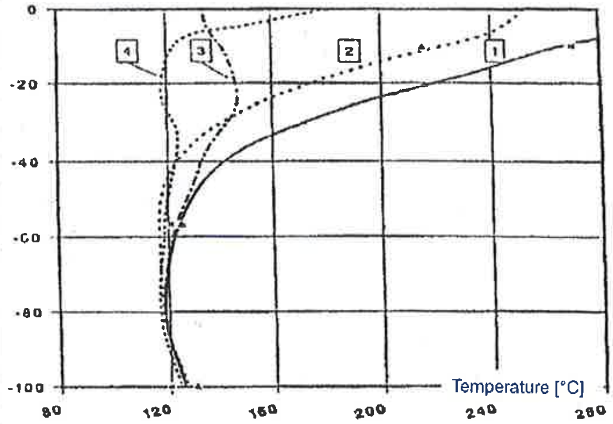


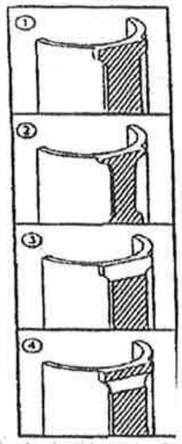
- IN HIGH PERFORMANCE ENGINES, WET LINERS SUPPORTED "ON THE MIDDLE" ARE COMMONLY ADOPTED.
- IN THIS SOLUTION, THE WET PART OF THE LINER HAS A LENGTH EQUAL TO 1/3 – 1/2 OF THE TOTAL LENGTH OF THE LINER.
- IN CLOSED DECK CYLINDER BLOCK, THE WATER JACKET LENGTH INFLUENCES THE TEMPERATURE DISTRIBUTION ALONG THE CYLINDER.



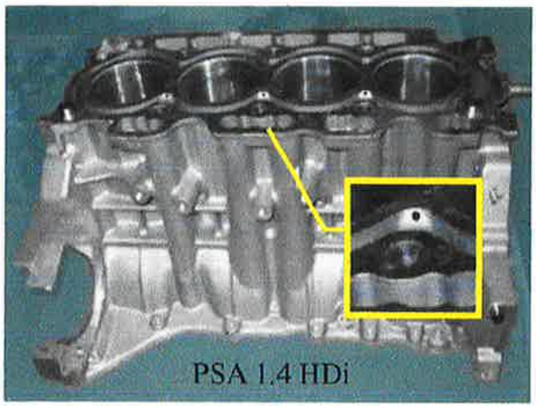


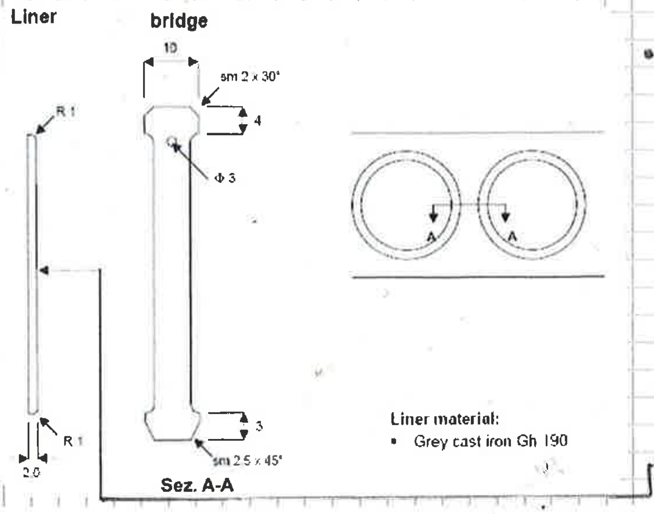
- THE TWINNED LINERS SOLUTION HAS TO GUARANTEE THAT THE "BRIDGE" BETWEEN THE LINERS HAS A THICKNESS EQUAL TO THE MINIMUM VALUE FOR ADEGUATE STRUCTURAL RESISTANCE.
- THE TEMPERATURE DISTRIBUTION ALONG THE LINER LENGTH DEPENDS ON THE BRIDGE GEOMETRY.





- THE TWINNED LINERS SOLUTION HAS THE ADVANTAGE OF COMPACTNESS.
- AS THE COOLING PATH BETWEEN THE LINERS IS ABSENT IT IS NOT POSSIBLE TO GUARANTEE A UNIFORM LOW TEMPERATURE IN THE TWINNING-AREA, WITH THE RISK TO GET HIGH DANGEROUS TEMPERATURE IN THE BRIDGES.
- TO ACHIEVE A GOOD COOLING EFFECT EVEN IN THE TWINNED LINERS CASE, A DEEP CFD STUDY IS NECESSARY.
- IN CASE OF HIGH TEMPERATURE IN THE BRIDGE, THE UPPER PART OF THE LINERS CAN BE PROVIDED BY A TRANSVERSAL CHANNEL (CROSS-DRILLING) TO COOL THE TWINNING-AREA.



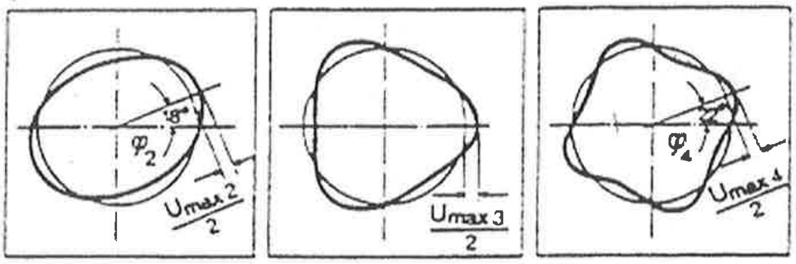


- SECTION AND MAIN DIMENSIONS OF THE BRIDGE ZONE (WITH CROSS-DRILLING). (LINER MATERIAL: GREY CAST-IRON GH190)

ROUGH CAST SURFACE (AS MAHLE SOLUTION)

- APPLYING THE FOURIER TRANSFORMATION TO THE COMPUTED TRANSVERSAL DEFORMATION OF THE LINER IT IS POSSIBLE TO IDENTIFY ITS DIFFERENT ORDERS:

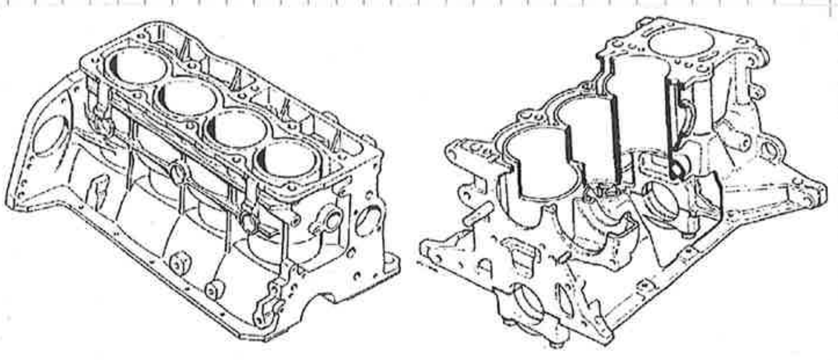
1ST ORDER (CIRCLE)
2nd ORDER (ELLIPSE)
3rd ORDER (TRI-LOBED SHAPE)
4th ORDER (FOUR-LOBED SHAPE)
...



A_2	B_2	$U_{max\,2}$	φ_2	A_3	B_3	$U_{max\,3}$	φ_3	A_4	B_4	$U_{max\,4}$	φ_4
4	3	10	18	4	0	8	0	0	4	8	22

- THE FOUR-LOBED SHAPE IS COMMONLY PRESENT IN BEDPLATE OF MEDIUM-HIGH STRESSED ENGINES, WHERE FOR EACH CYLINDER ARE PRESENT 4 FIXING SCREWS.

- THE MECHANICAL DEFORMATION OF THE LINER IS AFFECTED BY THE POSITION AND DEPTH OF THE THREADED HOLES FOR THE CYLINDER HEAD FIXING SCREWS.
- TO REDUCE THIS DEFORMATION, THE CYLINDER HEAD FIXING SCREWS ARE MOVED AS OUTSIDE AS POSSIBLE WITH RESPECT TO THE LINERS.
- CYLINDER BLOCKS WITH OPEN DECK SOLUTION ARE FAVOURED WITH RESPECT CLOSED DECK SOLUTION BECAUSE THE UPPER PART OF THE LINERS IS NOT LINKED TO THE EXTERNAL PANELS OF THE CYLINDER BLOCK WHERE THE SCREW HOLES ARE POSITIONED.



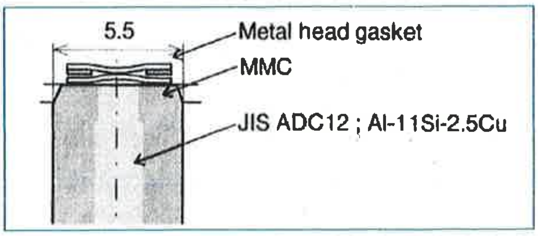
- L - BI-LAYER TYPE

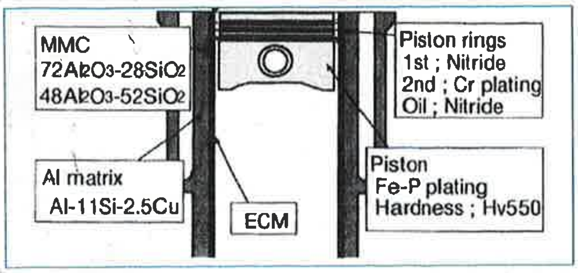
- AN EXAMPLE OF BI-LAYER LINERS ARE THE GOEDEL LINERS (BY FEDERAL MOGUL, 2006) MADE BY A LAMELLAR CAST-IRON IN THE INTERNAL WORKING SURFACE (TO IMPROVE MECHANICAL RESISTANCE) AND BY AN Al-ALLOY OUTSIDE (SUITABLE FOR COOLING AND ANCHORING TO THE BLOCK).
- THESE TWO MATERIALS HAVE SIMILAR THERMAL EXPANSION COEFFICIENTS.
- A TRANSITION ZONE IS PRESENT, WITH INTER-METALLIC BONDS THAT ENSURE A GOOD COHESION WITH THE CYLINDER BLOCK.

- L - MMC TYPE

- THE METAL MATRIX COMPOSITE (MMC) SOLUTION HAS BEEN PROPOSED BY TOYOTA (2000, ENGINE 2ZZ-GE, 1.8 l, 4 CYL.) ON Al-ALLOY CYLINDER BLOCK.
- THE GOAL WAS TO REALIZE VERY THIN BRIDGES WITH THICKNESS OF 5 [mm].
- A CASTING TECHNIQUE WITH FORMS FILLED IN LAMINAR FLOW REGIME WAS DEVELOPED TO ENSURE THE CONTROL OF TEMPERATURE, INJECTION VELOCITY AND DEGASSING PROCESS.

- SEALING BETWEEN CYLINDER HEAD AND CYLINDER BLOCK IS GUARANTEED WITH A MULTI-LAYER STEEL GASKET.
- THE 5 [mm] BRIDGE THICKNESS IS THE MINIMUM LIMIT VALUE THAT GUARANTEES A GOOD SEALING (FINAL LINER DIAMETER IS CHOSEN EQUAL TO 5.5 [mm]) ?
- THE MATRIX IS Al-ALLOY AlSi11Cu2.5 (NAMED JIS ADC12).
- THE PISTON SKIRT IS COATED WITH A Fe-P LAYER WITH HIGH HARDNESS AND GOOD MACHINABILITY.
- TO ENSURE GOOD LUBRICATION, AN ELECTRO-CHEMICAL MACHINING (ECM) IS NEEDED ON THE INTERNAL SURFACE OF THE LINER.





- MMC LINER COMPARED WITH OTHER LINERS (BRIDGE THICKNESS 5.5 [mm])

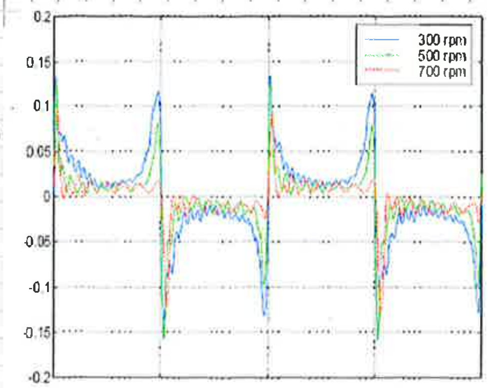
Cylinder block	Al-alloy				Cast iron
	MMC	Plasma deposition	Galvanic coating	Hypereutectic alloy	
Bridge temperature	B	B	B	B	D
Bridge stiffness	A	B	B	C	C
Bridge resistence	A	B	B	C	C
Minimum area for cylinder head gasket	A	B	B	B	B

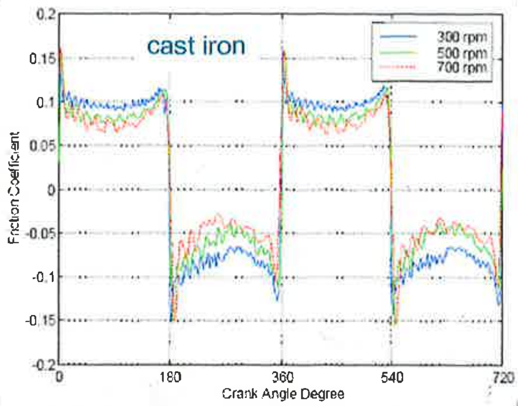
A = excellent; B = very good; C = good; D = not exhaustive

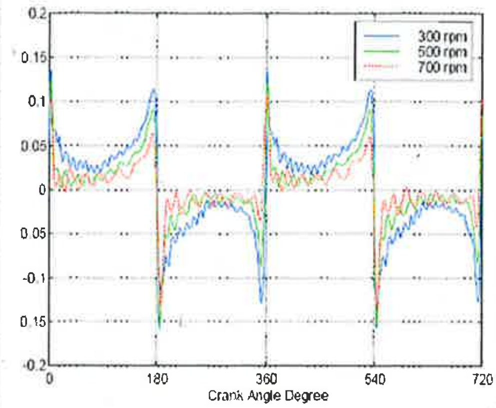
- L - PLASMA COATED TYPE

- PLASMA SPRAY PROCESS IS PROPOSED TO INCREASE MECHANICAL AND TRIBOLOGICAL CHARACTERISTICS.
- IT IS POSSIBLE TO OBTAIN EXCELLENT CHARACTERISTIC OF LUBRICANT RETENTION BY CONTROLLING THE DIMENSION OF THE COATING PARTICLES, THE PLASMA FLOW AND THE CURRENT.
- BN (BORON NITRIDE) HAS PARTICLES WITH SIMILAR CHARACTERISTIC OF SOLID LUBRICANT AND CAN BE EASILY USED IN PLASMA SPRAY PROCESS, SIGNIFICANTLY REDUCING THE FRICTION COEFFICIENT.
- COATING FRICTION COEFFICIENT INCREASES WITH TEMPERATURE.
- TWO PLASMA COATINGS WERE PROPOSED BY FORD (1999):
  - M1-P IS A COATING WITH IRON OXIDE Fe_2O_3 (LOW COST) CONTAINING STEEL PARTICLES (MEDIUM-LOW CONTENT OF CARBON), OBTAINED BY MEANS OF WATER-ATOMIZED PROCESS.
  - FFS-STANDARD IS A COATING WITH A MIX OF Ni-BN AND AISI 434 STEEL PARTICLES OBTAINED BY WATER-ATOMIZED PROCESS.

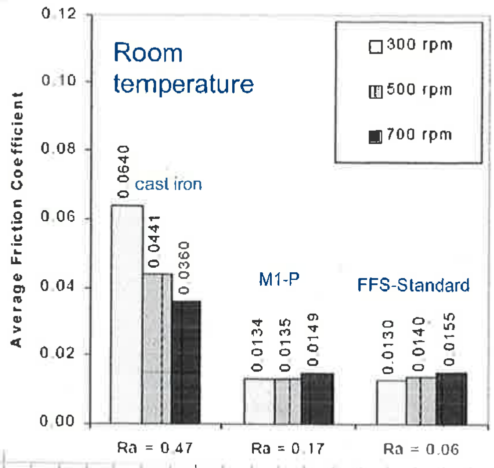
- ALSO AT HIGH TEMPERATURE (70[°C]), PLASMA COATING SHOWS HIGH CAPABILITY TO START A HYDRODYNAMIC TYPE LUBRICATION IF COMPARED WITH CAST IRON LINERS.

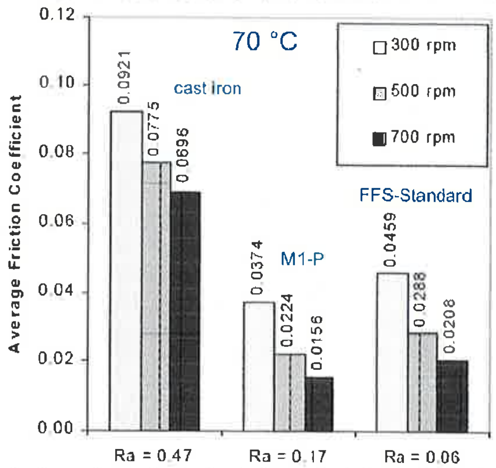






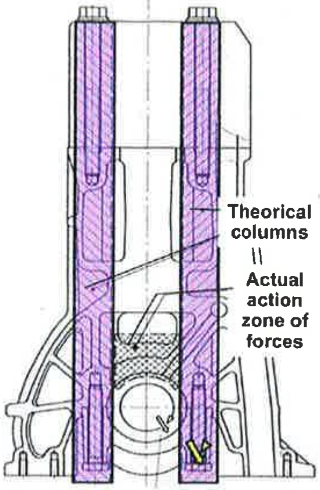
- M1-P AND FFS-STANDARD COATINGS SHOW A LOWER AVERAGE FRICTION COEFFICIENT WITH RESPECT CAST IRON.



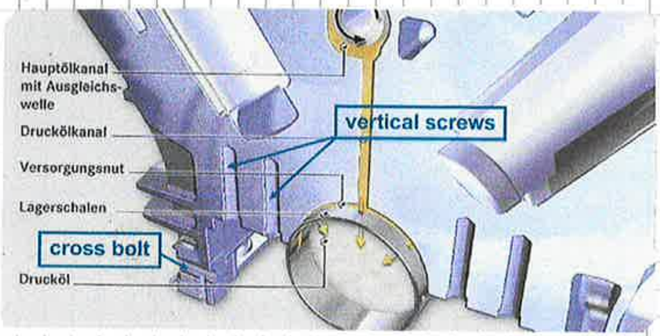


- AT ROOM TEMPERATURE, INCREASING THE ENGINE SPEED THE FRICTION COEFFICIENT DECREASES FOR CAST IRON AND SLIGHTLY INCREASES FOR M1-P AND FFS-STANDARD.
- AT HIGH TEMPERATURE (70[°C]) THE AVERAGE FRICTION COEFFICIENT DECREASES WITH THE INCREASING OF THE ENGINE SPEED FOR ALL PLASMA COATED LINERS.

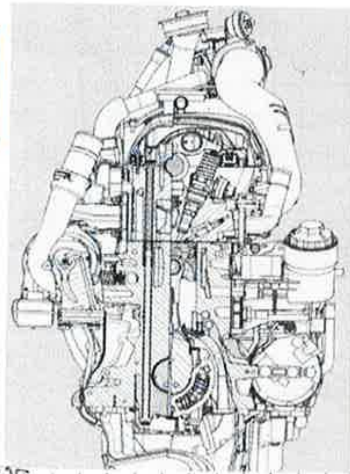
• FASTENING OF CYLINDER HEAD AND MAIN CAPS



- ACTIONS THAT GAS PRESSURE AND INERTIAL FORCES APPLY TO THE CYLINDER HEAD AND THE CRANK MECH. LOAD THE CYLINDER BLOCK.
- THE LOADED AREAS ARE TWO "THEORETICAL COLUMNS" THAT IDEALLY LINK THE HEADS OF THE FIXING SCREWS OF THE CYLINDER HEAD TO THE HEADS OF THE FIXING SCREWS OF THE MAIN CAPS.
- THE GOAL IS TO ALIGN AS MUCH AS POSSIBLE THE CYLINDER HEAD - CYLINDER BLOCK SCREWS WITH THE CRANKCASE - MAIN CAPS SCREWS, DEFINING THE CYLINDER BLOCK SO THAT ITS STRUCTURE FOLLOWS AS CLOSELY AS POSSIBLE THESE FORCE LINES.
- A COMMERCIAL SOLUTION WITH STUDS THAT FASTEN CYLINDER HEAD AND MAIN CAPS TO THE CYLINDER BLOCK IS FOR EXAMPLE THE VW TUAREG 2.5 TDI ENG.

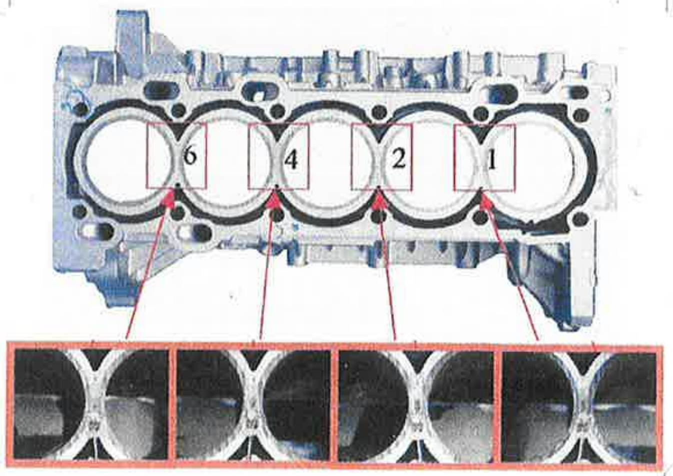


Al-alloy closed deck cylinder block with plasma coating liners (VW Tuareg 2.5 TDI)

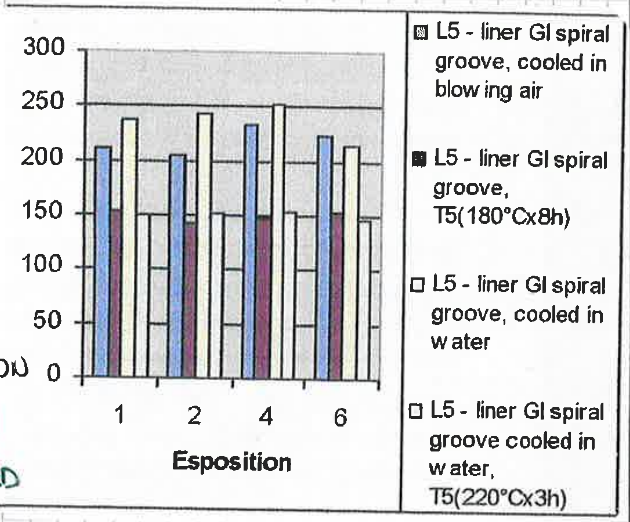


- THE LOADS DUE TO THE FASTENING OF THE CYLINDER HEAD TO THE CYLINDER BLOCK CAN PRODUCE ON THE MAIN CAPS REACTION FORCES WITH HORIZONTAL COMPONENTS OF HIGH VALUE.
- IN CASE OF Al-ALLOY CYLINDER BLOCK, THE SCREWS (USUALLY VERTICAL) WHICH FASTEN THE MAIN CAPS TO THE CYLINDER BLOCK STRUCTURE (WALLS) CANNOT TOLERATE HIGH FORCES, BEING FIXED IN Al, AND THEREFORE CANNOT AVOID THE POSSIBLE TRANSVERSAL SLIDING OF THE MAIN CAPS.
- TO REINFORCE THE MAIN CAPS AND THE CRANKCASE AREA, IRON INSERTS ARE COMBINED IN THE Al-ALLOY OF THE CRANKCASE.
- FOR EACH MAIN CAP, THE MOST USED FASTENING GEOMETRIES ARE:
  - 2 VERTICAL SCREWS - LOW STRESSED ENGINES.
  - 4 VERTICAL SCREWS (OR 2 VERTICAL + 2 TRANSVERSAL) - MEDIUM STRESSED ENG.
  - 4 VERTICAL SCREWS AND 2 TRANSVERSAL (CROSS BOLT) SCREWS OR BEDPLATE SOLUTION - HIGH STRESSED ENGINES.

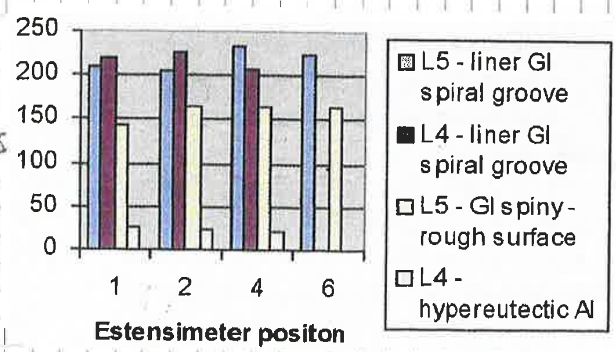
- DUAL GRID STRAIN GAUGES ARE GLUED ON THE BRIDGES BETWEEN ADJACENT LINERS AND THE DEFORMATION RELEASED DURING THE CYLINDER BLOCK CUTTING ARE MEASURED.



- RSEI - RESULTS



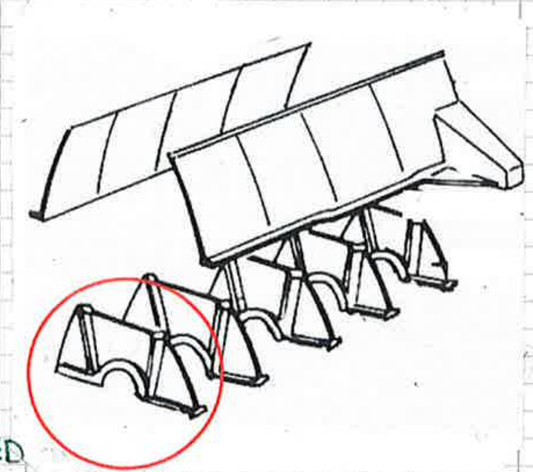
- THERMAL TREATMENT EFFECT ON GREY CAST IRON LINERS WITH SPIRAL GROOVE:
- ALL BRIDGES HAVE SIMILAR VALUES OF RESIDUAL STRESSES.
- FOR "AS-CAST" CYLINDER BLOCK, THE RESIDUAL STRESSES ARE NOT INFLUENCED (A LITTLE) BY THE TYPE OF COOLING AFTER EXTRACTION FROM THE MELTING FORM: 205-233 [MPa] IF COOLED IN AIR, 215-253 [MPa] IF COOLED IN WATER.
- BOTH THERMAL TREATMENTS T5, 180 [°C] 8 [h] AND 220 [°C] 3 [h], REDUCE THE RESIDUAL STRESSES TO 150 [MPa].



- LINER TYPE EFFECT:
- GREY CAST IRON LINERS WITH SPIRAL GROOVE PRESENT SIMILAR RESIDUAL STRESSES (205-233 [MPa] AND 207-226 [MPa]) FOR DIFFERENT NUMBER OF LINERS (5 OR 4)
- GREY CAST IRON LINERS WITH SPINY EXTERNAL SURFACE ARE LESS LOADED (143-166 [MPa]) WITH RESPECT TO GREY CAST IRON LINERS WITH SPIRAL GROOVE.
- HYPER EUTECTIC Al-ALLOY LINERS HAVE LOW RESIDUAL STRESSES (21-25 [MPa]).

- DESIGN GUIDELINES

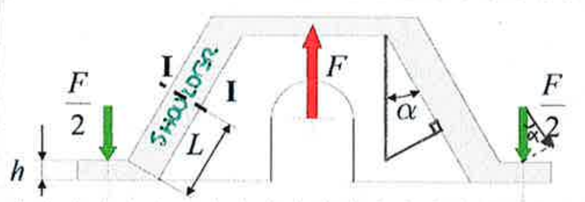
-DG- WALL



- WALLS ARE MADE BY CAST IRON OR STEEL WELDED PLATES AND ARE CONNECTED TO THE CYLINDER BLOCK SKIRTS BY MEANS OF SCREWS.
- WITH THE MAIN CAPS THEY FORM THE STRUCTURE THAT SUPPORTS THE CRANKSHAFT.
- THE CALCULATION MODEL CONSIDERS THE MAXIMUM RESULTANT FORCE (F) TRANSMITTED BY THE CRANKSHAFT TO THE WALL; DEPENDING ON THE INSTANT OF THE ENGINE CYCLE AND ON THE ENGINE SPIN SPEED, FORCE F CAN BE ASSUMED PROPORTIONAL TO $F_{g\,MAX}$ OR TO $F_{a\,MAX}$.
- AS WALLS CAN BE FIXED TO THE CYLINDER BLOCK BY MEANS OF SCREWS OR STUDS (THAT REACH UP TO THE CYLINDER HEAD) THE CALCULATION IS DIFFERENT.

- WHEN THE WALL IS FIXED TO THE CYLINDER BLOCK BY USING SCREWS ITS MOST STRESSED SECTION IS SECTION I-I (NEAR THE WALL INSPECTION WINDOW).

$$F = \begin{cases} F_{a\,MAX} = m_a \omega^2 R (1+\Lambda) \\ F_{g\,MAX} = P_{MAX} \dfrac{\pi D^2}{4} \end{cases}$$



- THE TENSILE FORCE (F_t) AND THE BENDING MOMENT (M_b) ACTING ON SECTION I-I ARE:

$$F_t^{I} = \frac{F}{2} \cos\alpha$$
$$M_b^{I} = \frac{F}{2} \sin\alpha \cdot L$$

- AND THE CORRESPONDING STRESSES ARE:

$$\sigma_t^{I} = \frac{F_t}{A_{I\text{-}I}} = \frac{(F/2)\cos\alpha}{A_{I\text{-}I}}$$

$$\sigma_b^{I} = \frac{M_b}{W_{I\text{-}I}} = \frac{(F/2)\sin\alpha \cdot L}{W_{I\text{-}I}}$$

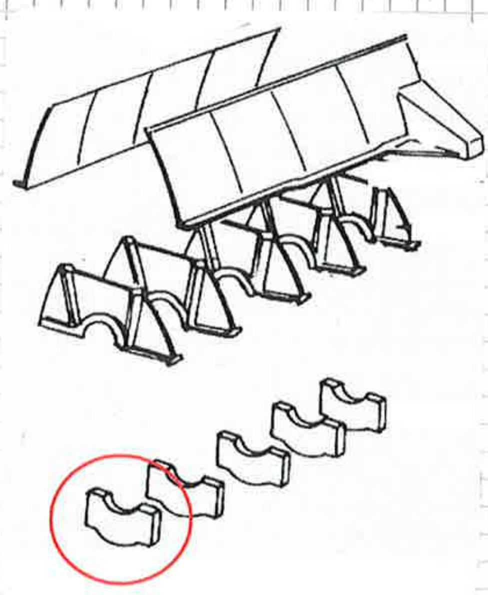
Q: WHERE THESE VALUES COME FROM? THEY AREN'T $R_{p0.2}/SF$, ARE THEY?

- THE RESULTANT STRESS IN SECTION I-I HAS TO BE LIMITED TO THE ADMISSIBLE MATERIAL STRESS:

$$\sigma_{EQ}^{I} = \sigma_t^{I} + \sigma_b^{I} \le \sigma_{ADM}$$

Material	σ_{adm} [MPa]
Cast iron, Al-alloy	20
Cast steel	40
Welded steel	60

- DG - MAIN CAP

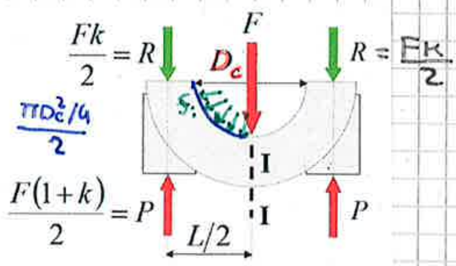


- THE LOAD ON THE MAIN CAPS DEPENDS ON THE ARRANGEMENT OF THE MAIN CAPS IN THE ENGINE ARCHITECTURE.
- IN GENERAL THE CRANKSHAFT IS MOUNTED BELOW THE WALLS AND THE MAIN CAPS ARE FIXED TO THE WALLS.
- THE MAXIMUM LOAD (F) THAT ACTS ON EACH MAIN CAP IS PROPORTIONAL TO THE MAXIMUM GAS FORCE:

$$F = F_{g\,MAX} = P_{MAX} \times \frac{\pi D^2}{4}$$

- ENDURANCE AND RELIABILITY SHOULD BE CHECKED BY CALCULATION OF THE BENDING STRESS IN THE MOST STRESSED SECTION OF THE MAIN CAP.
- THE MAIN CAP IS FIXED TO THE CORRESPONDING WALL BY MEANS OF SCREWS WITH LOAD P.
- THE CALCULATION MODEL CONSIDERS ALSO THE MAXIMUM RESULTANT FORCE (F) THAT COMES FROM THE CRANKSHAFT THAT LOADS THE MAIN CAP.
- THE REACTION FORCES (R) CAN BE COMPUTED:

$$R = \frac{Fk}{2}$$



- ON SECTION I-I ACTS THE BENDING MOMENT:

$$M_b^I = (P-R)\frac{L}{2} - \frac{1}{2} F \frac{D_c}{4} = \left(\frac{F(1+k)}{2} - \frac{Fk}{2}\right)\frac{L}{2} - \frac{F}{2}\frac{D_c}{4} =$$

$$= \frac{F}{4}\left(L - \frac{D_c}{2}\right) \quad \left(M_b^I = (P-R)\frac{L}{2} - \frac{1}{2}\sum_{i=1}^{\infty} S_i\, x_i \cong (P-R)\frac{L}{2} - \frac{1}{2} F \frac{\pi D_c^2/4}{\pi D_c} \doteq (P-R)\frac{L}{2} - \frac{1}{2} F \frac{D_c}{4} \right)$$

- THE CORRESPONDING BENDING STRESS HAS TO BE LIMITED TO THE ADMISSIBLE STRESS OF THE MAIN CAP MATERIAL:

$$\sigma_b^I = \frac{M_b^I}{W_{I-I}} = \frac{F(L - D_c/2)}{4\, W_{I-I}} \leq \sigma_{ADM} \quad \text{(STATIC VERIF.)}$$

(INDIPENDENTLY FROM SCREWS OR STUDS USE)

Material	σ_{adm} [MPa]
Cast iron	20
Cast steel	60
Forged steel	80

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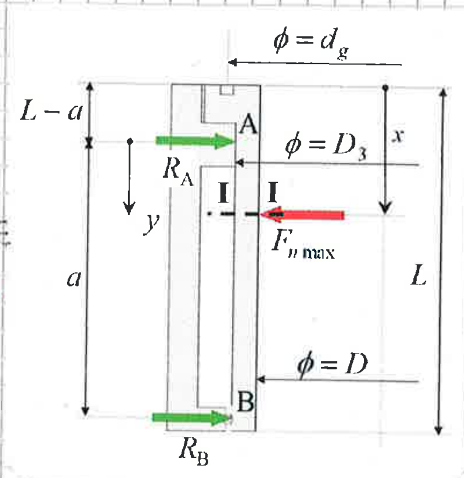
L2.2

- DG - LINER WALL



- THE LINER WALL HAS ALSO TO BE VERIFIED WITH RESPECT TO THE BENDING STRESS DUE TO FORCE F_n THAT LOADS ITS INTERNAL SURFACE.
- FROM CRANK MECHANISM DYNAMICS, IT IS POSSIBLE TO COMPUTE THE DISTANCE X (FROM TDC) WHERE THE FORCE F_n REACHES ITS MAXIMUM VALUE.
- FROM EQUILIBRIUM TO ROTATION AROUND POINT A, THE REACTION FORCE (R_B) IS:

$$R_B = F_{n\,MAX}\frac{y}{a} = F_{n\,MAX}\frac{x-(L-a)}{a} = F_{n\,MAX}\frac{(x-L+a)}{a}$$

- THE BENDING MOMENT IS:

$$M_b = R_B(L-x) = \frac{F_{n\,MAX}(x-L+a)}{a}\cdot(L-x)$$

- THE CORRESPONDING BENDING STRESS IS:

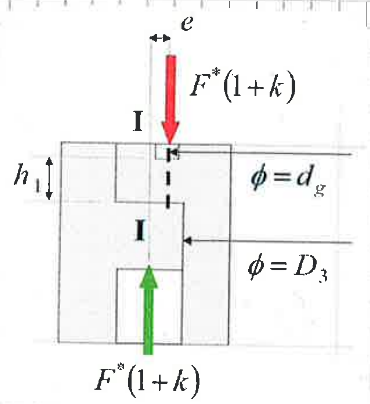
$$\sigma_b = \frac{M_b}{W_{I-I}} = \frac{M_b}{\frac{\pi(D_3^4-D^4)}{64\,D_3/2}} = \frac{32 D_3 F_{n\,MAX}(x-L+a)(L-x)}{a\pi(D_3^4-D^4)}$$

- THIS HAS TO BE COMPARED WITH THE MAXIMUM ADMISSIBLE STRESS OF THE LINER MATERIAL (I.E. CAST IRON LINER: 20 [MPa])

$$\sigma_b \le \sigma_{ADM}$$

L3

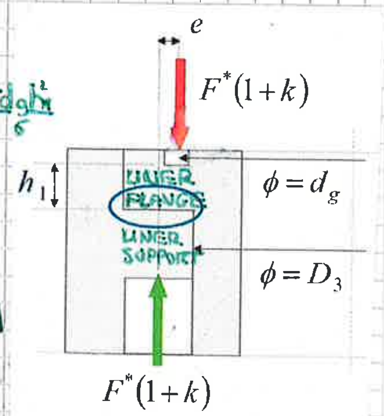
- DG - LINER FLANGE



- THE LINER FLANGE IS LOADED BY:
  - TIGHTENING FORCE PRODUCED BY CYL. H. SCREWS THROUGH THE CYL. H. GASKET.
  - BENDING GENERATED BY THE IMPERFECT ALIGNMENT BETWEEN THE GASKET HOUSING (MADE ON THE CYL. H) AND THE LINER SUPPORT.
- IN SECTION I-I SHEAR AND BENDING STRESS ARE:

$$\tau^I = \frac{F^*(1+k)}{A_{I-I}} = \frac{F^*(1+k)}{\pi d_g h_1} = P_{MAX}\frac{(1+k)d_g}{4h_1}$$

$$\sigma_b^I = \frac{F^*(1+k)\cdot e}{W_{I-I}} = \frac{F^*(1+k)e}{\pi d_g h_1^2/6} = \frac{3(1+k)d_g e}{2h_1^2} \quad , W_{I-I} = \frac{b\cdot h^2}{6} = \frac{\pi d_g h_1^2}{6}$$



- THE CONTACT PRESSURE WITH THE SUPPORT COLLAR IS:

$$P = \frac{F^*(1+k)}{A} = P_{MAX}\frac{(1+k)d_g^2}{(D_2^2-D_3^2)} \quad \left(F^* = P_{MAX}\frac{\pi d_g^2}{4}\ ;\ A = \frac{\pi(D_2^2-D_3^2)}{4}\right)$$

- THIS (P) HAS TO BE LOWER THAN A LIMIT VALUE: 80 [MPa]

$$P \le 80\,[MPa]$$

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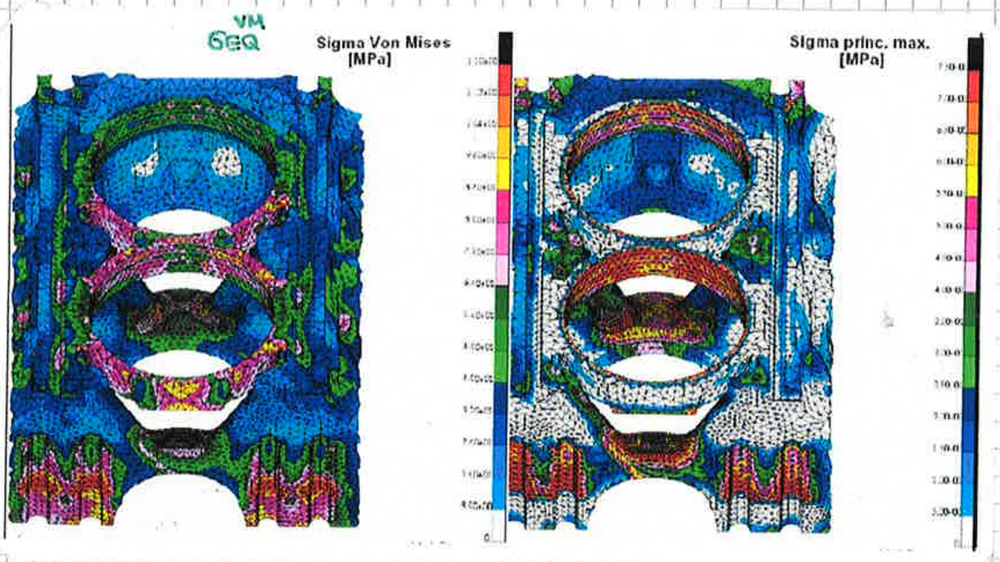
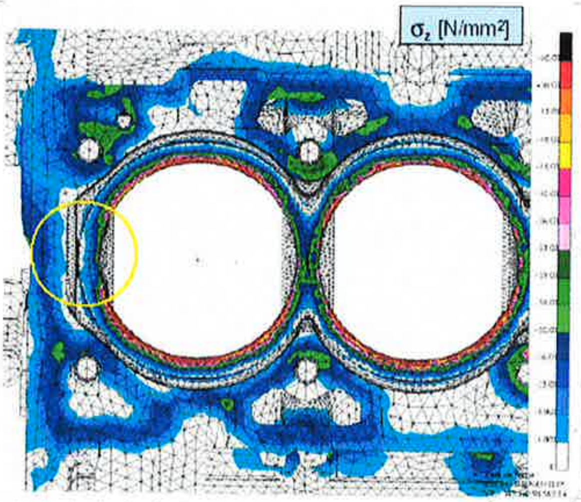
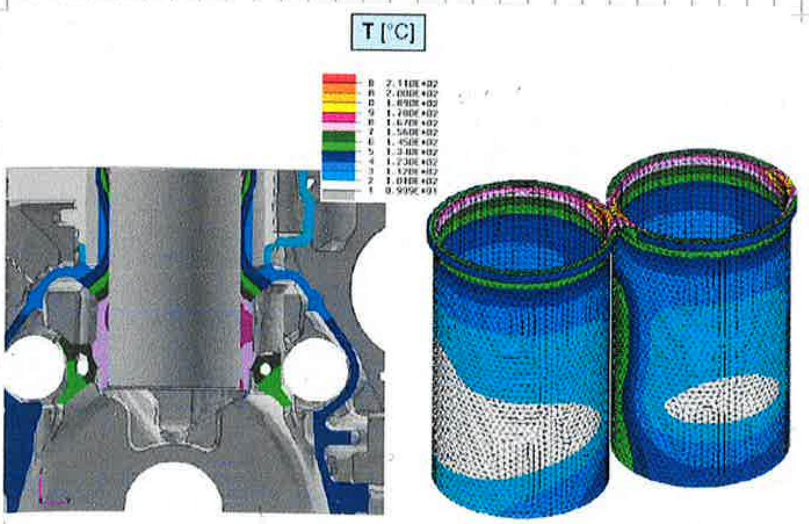
• STRUCTURAL ANALYSIS RESULTS (DYNAMIC LOADS)



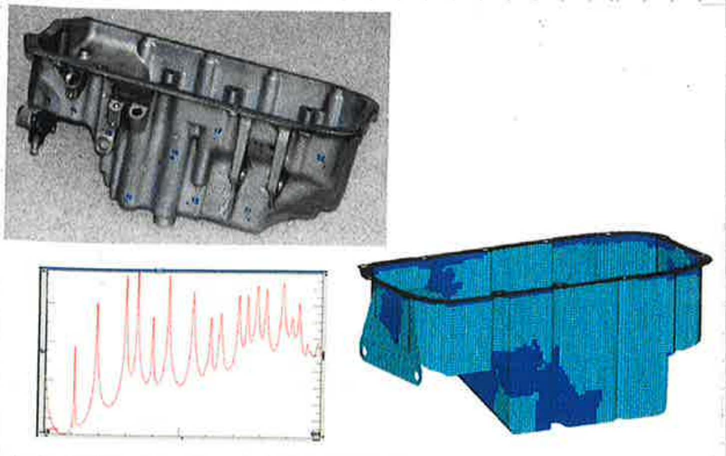
• STRESSES DUE TO SCREWS TIGHTENING.



• TEMPERATURE ANALYSIS



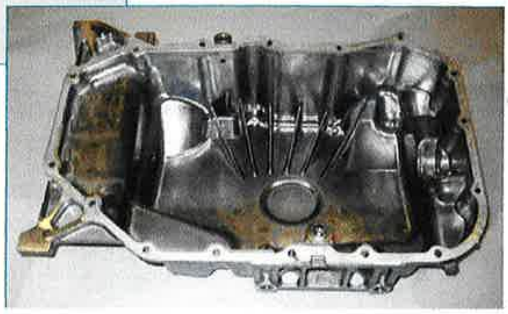
E9 OIL PAN



- INTRODUCTION
- EXAMPLES
- NUMERICAL OPTIMIZATION METHODOLOGY
  - NOM - PROPOSAL
  - NOM - CASE STUDY
  - NOM - VALIDATION
  - NOM - DESIGN APPLICATION

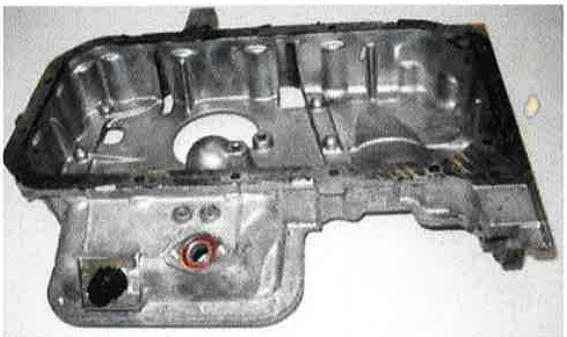
- EXAMPLES

Simple oil pan in steel sheet
(FIAT Fire 8v)

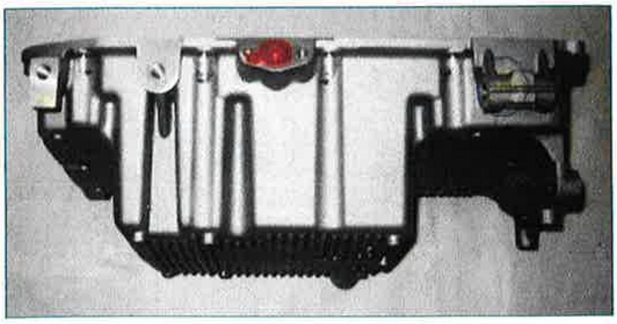


Simple oil pan in Al-alloy
(Honda 2.2 i-CTDi)

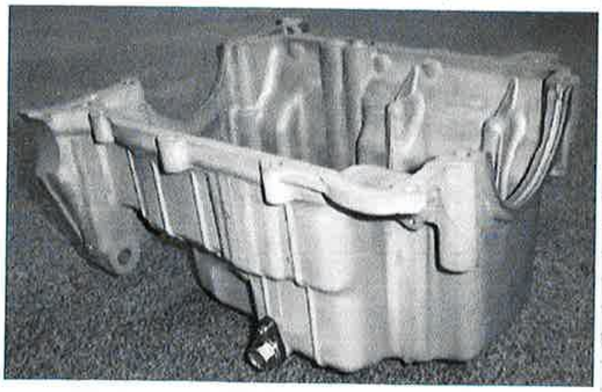
Simple oil pan in Al-alloy and double steel sheet
(Isuzu 1.t CDTI)



Structural oil pan
(FIAT 1.9 JTD)

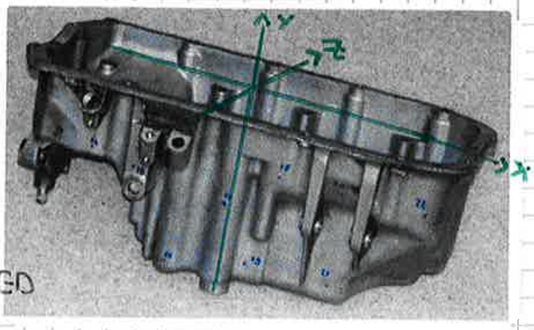


Structural Mg-alloy oil pan
(IMA system by Honda, 2000)

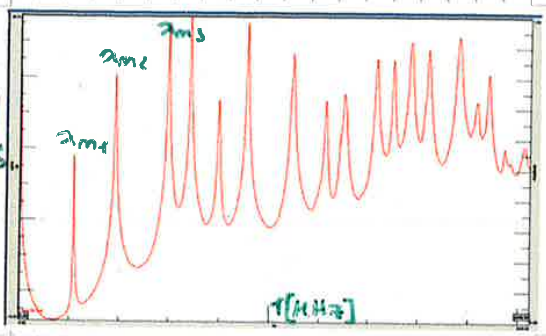


weight 2.5 kg, 35% less than Al-alloy solution

- NOM - CASE STUDY



- BY MEANS OF **EXPERIMENTAL MODAL ANALYSIS (EMA)** A COMMERCIAL **Al-ALLOY** OIL PAN IS CHARACTERIZED. **3.2 [kg] WEIGHT**
- A **MIMO** (MULTI INPUT MULTI OUTPUT) **MODAL ANALYSIS** IS USED (TEST-LAB DEDICATED SOFTWARE AND HARDWARE) WITH **IMPACT EXCITATION IN 3 DIRECTIONS** AND 51 MEASUREMENTS. (**INSTRUMENTED HAMMER**)

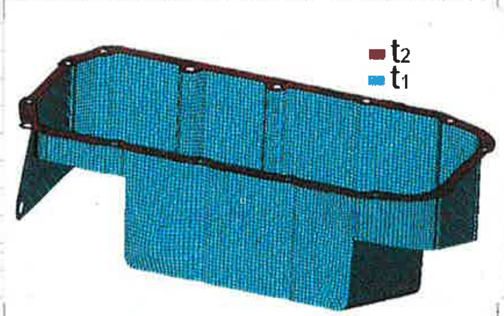


- THE **EXPERIMENTAL NATURAL FREQUENCIES** UP TO 1000 [Hz] ARE:

Mode	1	2	3	4	5	6
Freq. EMA [Hz]	218	382	597	681	796	912

- NOM - VALIDATION

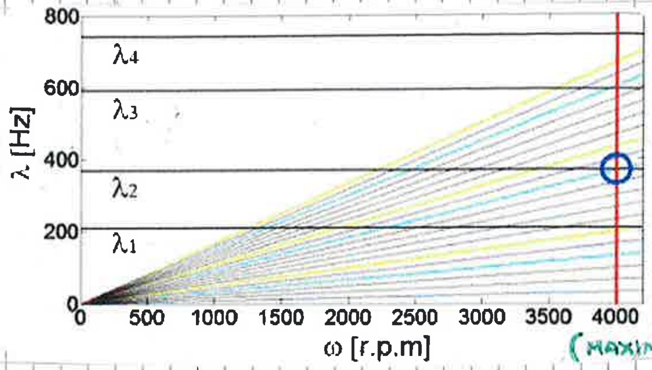
FIRST DESIGN MODEL: INITIAL FE MODEL



- DEFINITION OF A **SIMPLIFIED GEOMETRY**
- MATERIAL: **Al-ALLOY**
- THICKNESS: t_1 **MINIMUM THICKNESS ALLOWED BY CASTING** (2 [mm]) DEFINED AS "**DESIGN AREA**" (CYAN), t_2 **MINIMUM THICKNESS ALLOWED** FOR **BOLT CONNECTION** (7 [mm]) ALONG THE OIL **PAN EDGE** DEFINED AS "**NON-DESIGN AREA**" (RED).
- **NMA**: THE **COMPUTED FIRST FOUR NATURAL FREQUENCIES** ARE **LOWER** THAN THE **EXPERIMENTAL ONES** THEREFORE A **FIRST OPTIMIZATION** IS **NEEDED**.

Mode	1	2	3	4
Freq. EMA [Hz]	218	382	597	681
NMA [Hz]	176	278	482	620

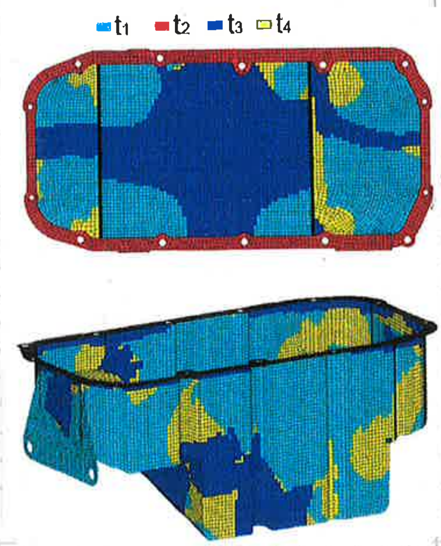
- THE COMPUTED NATURAL FREQUENCIES MUST BE PLACED ON THE CAMPBELL DIAGRAM IN ORDER TO CHOOSE THE FREQUENCY TO MAXIMIZE DURING THE SECOND OPTIMIZATION.



λ_2 FIRST NATURAL FREQUENCY NEAR TO RESONANT CONDITION → WE HAVE TO INCREASE IT.

- SECOND NATURAL FREQUENCY (λ_2) INTERSECTS THE 11th ENGINE HARMONIC AT THE MAXIMUM ENGINE WORKING SPEED, THEREFORE THE SECOND OPTIMIZATION WILL HAVE TO MAXIMIZE THE SECOND NATURAL FREQUENCY (λ_2).

SECOND OPTIMIZATION → THIRD DESIGN MODEL



- THICKNESS: t_1 2 [mm] "DESIGN AREA" (CYAN),
  t_2 7 [mm] "NON-DESIGN AREA" (RED),
  t_3 5 [mm] "NON-DESIGN AREA" (BLUE).
- NEW OPTIMIZED ZONES WITH t_4 5 [mm] (YELLOW)
- NMA

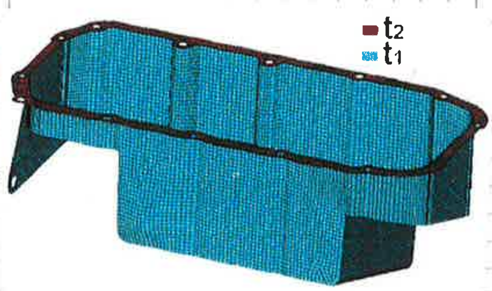
Mode	1	2	3	4
Freq. EMA [Hz]	218	382	597	681
NMA [Hz]	206 -6%	433 +13%	651 +9%	803 +18%

- WITH RESPECT TO THE EXPERIMENTAL NATURAL FREQUENCIES:

1st: −6%
2nd: +13%
3rd: +9%
4th: +18% → TARGET ACHIEVED (BEST OPTIMIZED DESIGN)

- NON-DESIGN APPLICATION

FIRST MODEL: INITIAL FE MODEL



- MATERIAL: PA66 COMPOSITE (POLYMERIC MAT.) (LONG GLASS FIBER REINFORCED)
- THICKNESS: t1 MINIMUM THICKNESS ACCORDING TO MANUFACTURING TECHNOLOGY (5 [mm]) DEFINED AS "DESIGN AREA" (CYAN), t2 THICKNESS ALLOWED FOR BOLT CONNECTION (10 [mm]) ALONG THE OIL PAN EDGE DEFINED AS "NON-DESIGN AREA" (RED).
- NMA: THE COMPUTED FIRST FOUR NATURAL FREQ. ARE LOWER THAN THE EXPERIMENTAL ONES THEREFORE A FIRST OPTIMIZATION IS NEEDED.

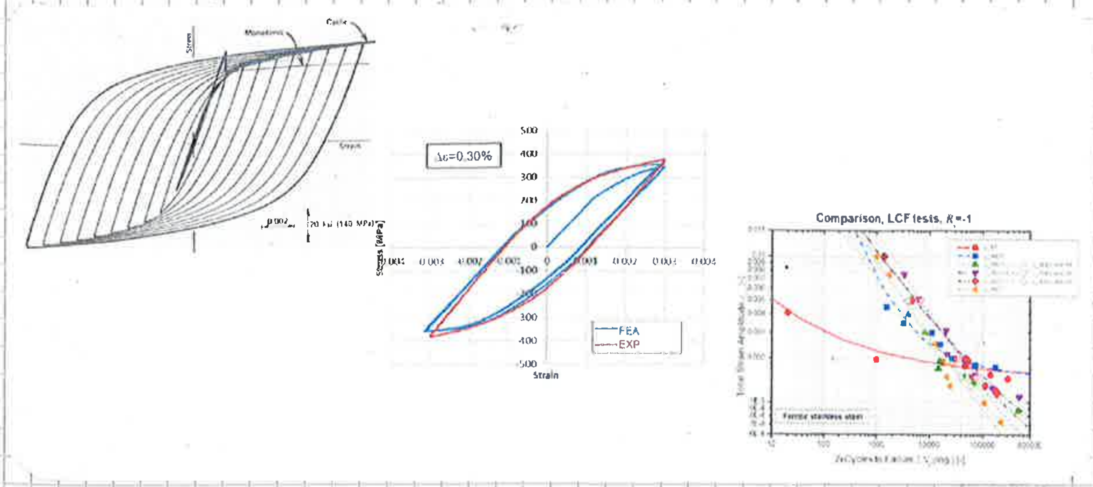
Mode	1	2	3	4
Freq. EMA [Hz]	218	382	597	681
NMA [Hz]	151	293	470	592

FIRST OPTIMIZATION

- TARGET: INCREASE THE NATURAL FREQUENCIES
- CONSTRAINTS: ACT ONLY CHANGING THE MATERIAL DENSITY IN THE "DESIGN AREA" OF THE OIL PAN (THAT CORRESPONDS TO MODIFY THE THICKNESS)
- NUMERICAL MODAL ANALYSIS (NMA) ON FE MODELS WITH DIFFERENT "MATERIAL DENSITY" WHICH CORRESPONDS TO DIFFERENT STRUCTURAL THICKNESS.
- A GOOD COMPROMISE BETWEEN WEIGHT SAVING AND FREQUENCY REQUEST IS OBTAINED WITH A DENSITY CHANGE OF 50%: MINIMUM THICKNESS 5 [mm] (DARK BLUE AREAS, t1) AND MAXIMUM THICKNESS 7 [mm] (RED AREAS, t3).

Density change 50%

E10 THERMO-MECHANICAL FATIGUE



- INTRODUCTION
- MATERIAL RESPONSE
  - MR - HYSTERESIS LOOP IN ISOTHERMAL FATIGUE
  - MR - HYSTERESIS LOOP IN THERMO-MECHANICAL FATIGUE
- TMF DAMAGE
  - TMF D - TEMPERATURE DEPENDENCE OF MATERIAL PROPERTIES
  - TMF D - MECHANICAL FATIGUE
  - TMF D - MECHANICAL FATIGUE, BAUSHINGER EFFECT
  - TMF D - MECHANICAL FATIGUE, CYCLIC HARDENING / SOFTENING
  - TMF D - MECHANICAL FATIGUE, SHAKEDOWN EFFECT
  - TMF D - MECHANICAL FATIGUE, RATCHETING
  - TMF D - CREEP AND OXIDATION
  - TMF D - FATIGUE, CREEP AND OXIDATION CRACKS
- TMF RESIDUAL LIFE ESTIMATION
- MATERIAL CONSTITUTIVE LAWS
- DAMAGE MODELS
- UNIAXIAL DAMAGE MODELS
  - UDM - BASQUIN - MANSON - COFFIN MODEL
  - UDM - NEU - SEHITOGLU MODEL
  - UDM - CHABOCHE MODEL
  - UDM - BMW MODEL
- ENERGY-TYPE DAMAGE MODELS
  - ETDM - SKELTON MODEL
- UNIAXIAL AND ENERGY-TYPE MODELS COMPARISON
- MULTIAXIAL DAMAGE MODELS
  - MDM - VON MISES MODEL
  - MDM - ASME CODE MODEL
  - MDM - SONSINO - GRUBISIC MODEL
  - MDM - KANDIL - BROWN - MILLER MODEL
  - MDM - FATEMI - SOCIE MODEL

- THE CONSTITUTIVE MODELS OF MATERIAL (REPRESENTING THE MATERIAL BEHAVIOUR) HAS TO CONTAIN MECHANICAL PARAMETERS THAT ARE FUNCTION OF THE TEMPERATURE.
- ANOTHER EFFECT OF THE THERMAL FIELD IS THE POSSIBLE ACTIVATION OF CREEP AND OXIDATION PHENOMENA THAT MUST BE TAKEN INTO ACCOUNT BY THE RESIDUAL LIFE ESTIMATION MODELS. (CREEP: PROGRESSIVE STRAIN ACCUMUL. UNDER CONST. LOAD WHEN $T \geq \frac{1}{3} T_M$)
- ACCORDING TO ASTM E2368-04E1 STANDARD, TMF CYCLE IS A LOADING CONDITION IN WHICH TEMPERATURE AND STRAIN FIELDS ACT SIMULTANEOUSLY ON A SPECIMEN/COMPONENT, VARY AND ARE CONTROLLED SEPARATELY AND INDIPENDENTLY EACH OTHER. (OXIDATION: WHEN ENV. IS CHARACT. BY AGGRESSIVE GAS) [EXHAUST MANIFOLD IS SUBJECTED TO BOTH]
- THE MAIN GOALS [2(3)] OF TMF ANALYSIS ARE:
  1 - PREDICTION OF THE MATERIAL BEHAVIOUR UNDER SIMULTANEOUS VARIATION OF THERMAL AND MECHANICAL LOADS.
  2 - RESEARCH AND DEFINITION OF THE MAIN FACTORS INVOLVED IN THE DAMAGE MECHANISMS WHICH CAUSE THE COMPONENT FAILURE.
  3 - ESTIMATION OF THE COMPONENT RESIDUAL LIFE.
- TO ACHIEVE THE GOALS OF TMF ANALYSIS SEVERAL EXPERIMENTAL TESTS [3(3)] HAVE TO BE PERFORMED ON THE COMPONENT MATERIAL:
  1 - CONSTANT TEMPERATURE AND VARIABLE MECHANICAL LOAD (ISOTHERMAL MECHANICAL FATIGUE TEST)
  2 - VARIABLE TEMPERATURE AND CONSTANT MECHANICAL LOAD (THERMAL FATIGUE TEST)
  3 - VARIABLE TEMPERATURE AND VARIABLE MECHANICAL LOAD (THERMO-MECHANICAL FATIGUE TEST).
- THE PRINCIPAL EXPERIMENTAL DEVICES [4(2)] ARE:
  1 - VIBROFORE (HCF TESTS, ROOM TEMPERATURE RT OR HIGH ISOTHERMAL T)
  2 - SERVO HYDRAULIC MACHINE (LCF TESTS, ROOM TEMPERATURE RT OR HIGH ISOTHERMAL T. TMF TESTS).
- ALL TESTS CAN BE PERFORMED UNDER TWO DIFFERENT CONTROL MODES [5(2)]:
  1 - STRAIN-CONTROLLED TESTS (IMPOSITION OF THE TOTAL STRAIN RANGE $\Delta\varepsilon_{tot}$)
  2 - STRESS-CONTROLLED TESTS (IMPOSITION OF STRESS RANGE $\Delta\sigma$, WITH ZERO σ_m OR NOT).

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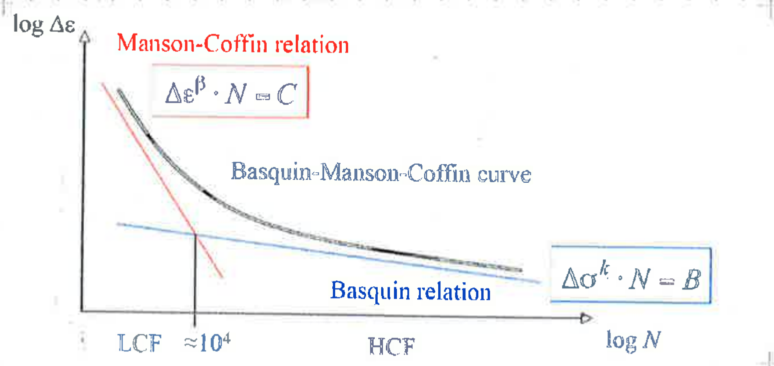
- IN HCF TESTS, STRESS-CONTROLLED OR STRAIN-CONTROLLED TESTS GIVE ALMOST THE SAME RESULTS AND THE BASQUIN RELATION IS USED TO CORRELATE THE STRESS AND THE NUMBER OF CYCLES:

$\Delta\sigma^k \cdot N = B$ (BASQUIN RELATION, HCF) (HCF PARAMETERS: σ_a, σ_m)

- IN LCF TESTS, VERY DIFFERENT RESULTS ARE ACHIEVED BY STRESS-CONTR. AND STRAIN-CONTROLLED TESTS.
- LCF TESTS ARE COMMONLY CARRIED OUT AS STRAIN-CONTROLLED TESTS (DISPLACEMENT-CONTROLLED) BECAUSE THEY BETTER CHARACTERIZE THE MATERIAL BEHAVIOUR COMPARED WITH THE STRESS-CONTROLLED TESTS.
- THE MANSON-COFFIN RELATION IS USED TO CORRELATE STRAIN AND THE NUMBER OF CYCLES:

$\Delta\varepsilon^\beta N = C$ (MANSON-COFFIN RELATION, LCF) (LCF PARAMETERS: $\varepsilon_a, \varepsilon_m$)

- TO ANALYZE BOTH HCF AND LCF FIELDS, THE EXPERIMENTAL DATA ARE REPORTED ON A $\Delta\varepsilon$-N GRAPH (LOG SCALES, BASQUIN-MANSON-COFFIN CURVE) INSTEAD OF ON THE $\Delta\sigma$-N GRAPH, TYPICAL FOR HCF CASE.
- A QUITE SIMILAR GRAPH IS USED ALSO IN CASE OF TMF INVESTIGATION.



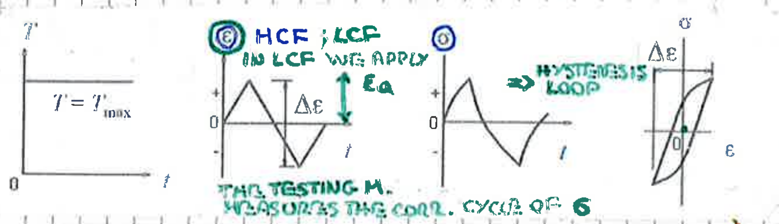
- THE TOTAL STRAIN RANGE IS THE SUM OF THERMAL STRAIN AND MECHANICAL STR.

$\Delta\varepsilon_{TOT} = \Delta\varepsilon_{th} + \Delta\varepsilon_{MECH}$

• MATERIAL RESPONSE

– MR – HYSTERESIS LOOP IN ISOTHERMAL FATIGUE

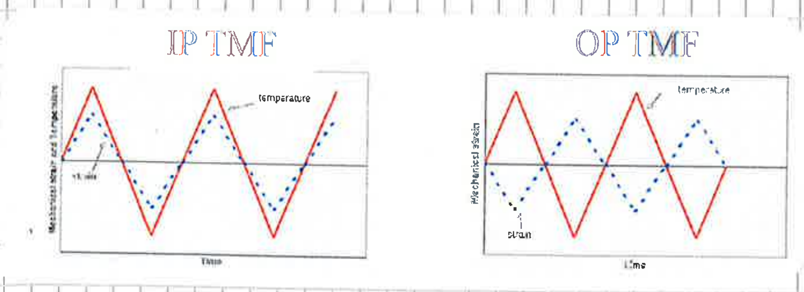
- IN ISOTHERMAL FATIGUE TEST, THE TESTING TEMPERATURE T IS CONSTANT AND GENERALLY EQUAL TO THE MAXIMUM WORKING TEMPERATURE T_{MAX}.
- THE STRAIN RANGE ($\Delta\varepsilon$) IS IMPOSED AND THE RESULTING STRESS IS ACQUIRED.
- COMBINING TOGETHER THE STRAIN AND THE STRESS RANGES, THE HYSTERESIS LOOP IS OBTAINED (EG GRAPH).
- IF THE IMPOSED STRAIN IS SYMMETRIC (E.G. ± 0.5%) USUALLY THE HYSTERESIS LOOP IS ALSO SYMMETRIC WITH RESPECT TO THE ORIGIN OF THE EG AXES.



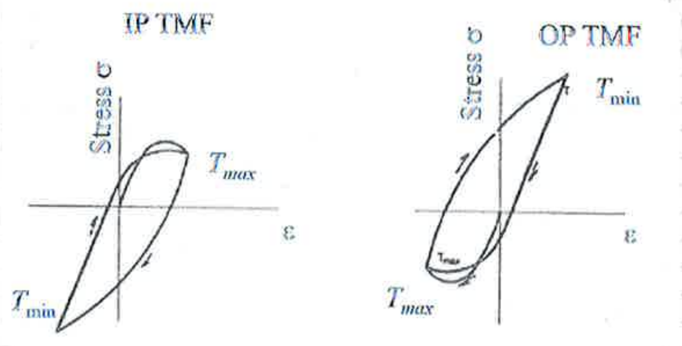
WHEN WE STUDY ANY KIND OF FATIGUE (IN PARTICULAR LCF AND THEN TMF THAT USE SAME KIND OF MODEL) FIRST OF ALL WE HAVE TO CHARACTERIZE THE MATERIAL UNDER SPECIFIC EXPERIMENTAL TESTS (IN WHICH WE CONTROL STRESS OR STRAIN).

– MR – HYSTERESIS LOOP IN THERMO-MECHANICAL FATIGUE

- IN THERMO-MECHANICAL FATIGUE TEST, THE TESTING TEMPERATURE T VARIES CYCLICALLY BETWEEN TWO FIXED VALUES.
- TEMPERATURE CYCLES ARE SUPERIMPOSED TO THE STRAIN CYCLES WHICH ARE INDEPENDENT FROM THE TEMPERATURE.
- DEPENDING ON THE PHASE BETWEEN THE MECHANICAL AND THE THERMAL CYCLES, TMF TESTS ARE IN-PHASE TESTS OR OUT-OF-PHASE TMF TESTS.
- IN IP TMF TEST MAXIMUM TEMPERATURE AND MAXIMUM STRAIN OCCUR AT THE SAME INSTANT; IN OP TMF TEST TEMPERATURE IS MAXIMUM WHEN DEFORMATION IS MINIMUM AND VICE VERSA.
- IN IP TMF THE HYSTERESIS LOOP USUALLY SHOWS A COMPRESSIVE MEAN STRESS
  IN OP TMF THE HYSTERESIS LOOP USUALLY SHOWS A TENSILE MEAN STRESS.



- IN OP TMF THE MATERIAL IS COMPRESSED WHEN T IS MAXIMUM AND STRETCHED WHEN T IS MINIMUM; THE MEAN STRESS (σ_m) IS THEN POSITIVE (TENSILE CONDITION).
- IN ADDITION, THE OP CONDITION IS MORE SENSITIVE THAN THE IP CONDITION TO THE OXIDATION EFFECT BECAUSE THE OXIDE LAYER IS FORMED AT HIGH TEMPERATURE DURING THE PHASE OF MECHANICAL COMPRESSION.
- WHEN THE TEMPERATURE DECREASES THE OXIDE LAYER BECOMES BRITTLE AND THE SUBSEQUENT PHASE OF MECHANICAL TRACTION RAPIDLY BREAKS THE OXIDIZED LAYER EXPOSING "NEW" MATERIAL TO THE SUBSEQUENT OXIDATION.
- THE ACTUAL TREND OF THE MATERIAL RESPONSE IN IP TMF AND OP TMF IS NOT THE ONE PREVIOUSLY SCHEMATICALLY REPORTED, BUT IT SHOWS AN HYSTERESIS LOOP SHIFTED TOWARD THE COMPRESSION FIELD (IF IP) OR SHIFTED TOWARD THE TENSILE FIELD (IF OP).

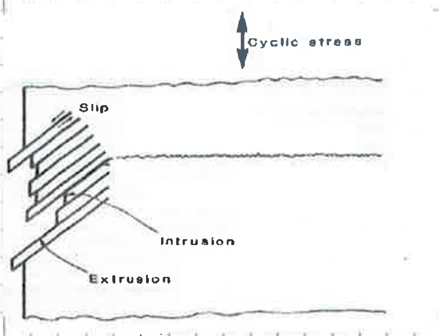


- ON THE BASIS OF THE PREVIOUS CONSIDERATIONS IT IS EVIDENT THAT OP TMF IS THE MOST DANGEROUS CONDITION FOR THE MATERIAL FAILURE. (OP TMF MORE DANGEROUS BECAUSE σ_m^+ IS MORE DANGEROUS THAN σ_m^-)

⇒ THE EXPERIMENTAL CHARACTERIZATION, FOR METALLIC MATERIALS, TO CHARACTERIZE THE MATERIAL BEHAVIOUR UNDER THERMO-MECHANICAL FATIGUE IS ALWAYS PERFORMED USING AN OP TMF INVESTIGATION.

- TMF D - MECHANICAL FATIGUE

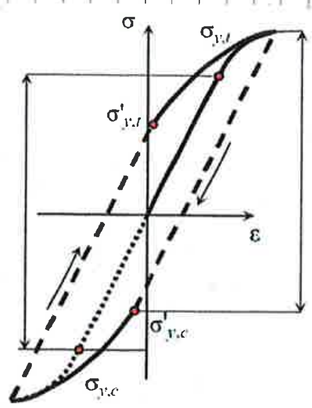
- THE MECHANICAL FATIGUE DAMAGE IS DUE TO THE HYSTERESIS OF THE MATERIAL WHICH IS CAUSED BY THE LOADING CONDITIONS.
- THE CRACK NUCLEATION AND PROPAGATION ARE CAUSED BY THE MATERIAL SLIDING ALONG CRYSTALLINE PLANES.



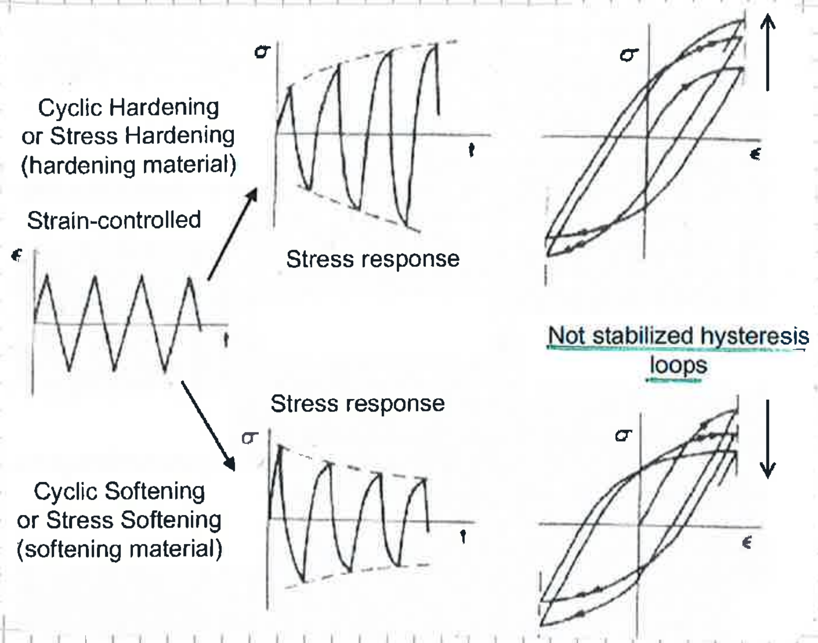
- INTRUSIONS AND EXTRUSIONS GROW ON THE SURFACE OF THE MATERIAL AND REPRESENT POTENTIAL NUCLEATION SITE FOR THE FATIGUE CRACK.
- THE MECHANICAL FATIGUE DAMAGE IS ENHANCED IF THE MATERIAL SHOWS PECULIAR BEHAVIOURS (SUCH AS BAUSCHINGER EFFECT (1.1), CYCLIC HARDENING OR CYCLIC SOFTENING (1.2), SHAKEDOWN EFFECT (1.3), AND RATCHETING (1.4)).

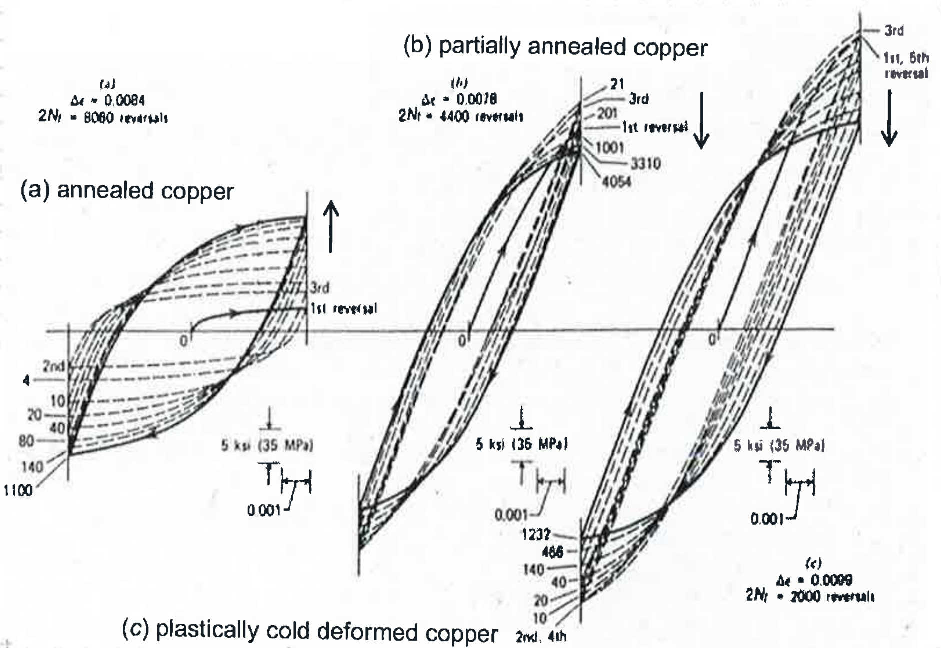
- TMF D - MECHANICAL FATIGUE, BAUSHINGER EFFECT (1.1)

- BAUSHINGER EFFECT IS A MATERIAL BEHAVIOUR RELATED TO THE FIRST LOADING CYCLE: THE YIELD STRESS OF THE MATERIAL AT THE FIRST LOAD APPLICATION IS USUALLY HIGHER THAN THE YIELD STRESS SHOWN BY THE MATERIAL DURING THE SUCCESSIVE HYSTERESIS CYCLES.
- THE BAUSHINGER EFFECT CAN BE THEN DESCRIBED AS A DECREASE OF THE COMPRESSIVE ELASTIC LIMIT AFTER THAT THE VIRGIN MATERIAL, I.E. LOADED FOR THE FIRST TIME, HAS BEEN STRESSED BEYOND ITS YIELD STRENGTH.



- PHYSICAL CAUSES OF THE BAUSHINGER EFFECT ARE THE PERMANENT DEFORMATIONS AT CRYSTALLINE LEVEL, THAT GENERATE MATERIAL ANISOTROPIES.
- AT MACROSCOPIC LEVEL, THESE ANISOTROPIES ARE MANIFESTED WITH THE DECREASE OF THE ELASTIC LIMIT AFTER THE APPLICATION OF PERMANENT PLASTIC DEFORMATIONS. [$\sigma_{y,t} > \sigma'_{y,t}$; $|\sigma_{y,c}| > |\sigma'_{y,c}|$]


Cyclic Hardening
or Stress Hardening
(hardening material)
Strain-controlled
Stress response
Not stabilized hysteresis loops
Stress response
Cyclic Softening
or Stress Softening
(softening material)

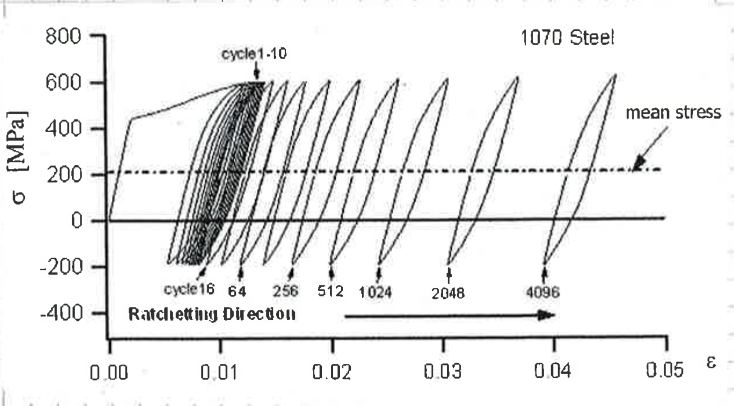

(b) partially annealed copper
(a)
Δε = 0.0084
2Nf = 8080 reversals
(b)
Δε = 0.0078
2Nf = 4400 reversals
(a) annealed copper
3rd
1st reversal
5 ksi (35 MPa)
0.001
(c)
Δε = 0.0099
2Nf = 2000 reversals
(c) plastically cold deformed copper

1.9

- TMF D - MECHANICAL FATIGUE, RATCHETTING (/RATCHETING)

- DEVELOPING STRESS-CONTROLLED TESTS INSTEAD OF STRAIN-CONTROLLED TESTS, THE RESPONSE OF THE MATERIAL CHANGES. IN PARTICULAR, IN NOT SYMMETRICAL STRESS-CONTROLLED TESTS (I.E. WITH UNSYMMETRICAL LOADING CYCLES WITH NON-ZERO MEAN STRESS) WHICH LEAD TO EXCEED THE MATERIAL YIELD STRENGTH, IT CAN BE OBSERVED A SLOW AND PROGRESSIVE ACCUMULATION OF PLASTIC DEFORMATION THAT CAN CONTINUE TO PROGRESS (INFINITE RATCHETTING) OR TO GET TO STABILIZE (FINITE R.)
- RATCHETTING IS A NOT STABLE MATERIAL BEHAVIOUR: HYSTERESIS LOOPS ARE NOT CLOSED BUT ARE CONSTANTLY MOVING ON THE σε PLANE TOWARDS INCREASING VALUES OF DEFORMATION, IN THE DIRECTION OF THE APPLIED MEAN STRESS.
- MATERIALS THAT SOFTEN CYCLICALLY SHOW A PROGRESSIVE INCREASE OF THE MEAN DEFORMATION UNTIL FAILURE.
- ON THE CONTRARY, MATERIALS THAT HARDEN CYCLICALLY SHOW A PROGRESSIVE DECREASE OF THE MEAN DEFORMATION (RATCHETTING RATE TENDS TO DECREASE) TO REACH, AFTER A CERTAIN NUMBER OF CYCLES, A STABILIZED STATE, THAT IS A CLOSED HYSTERESIS LOOP (RATCHETTING RATE GOES TO ZERO)
- RATCHETING IS THEN A PROGRESSIVE PLASTIC STRAIN ACCUMULATION WHICH VERIFIES AT EVERY CYCLE AND THAT CAN STABILIZE OR NOT:

$$\varepsilon_{ACC} = \int \varepsilon_{PL}\, dt$$



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2.4
- EMPIRICAL MODELS PROVIDE SIMPLE FORMULAE THAT CORRELATE CREEP RATE, STRESS, TEMPERATURE AND TIME FOR THE ESTIMATION OF THE MATERIAL BEHAVIOUR; THESE MODELS ARE USEFUL IN THE EARLY STEP OF DESIGN PROCESS AND ARE A VERY COMMON APPROACH IN APPLIED RESEARCH.
- PHYSICAL CAUSES OF CREEP ARE RELATED TO ATOMS DIFFUSION IN CRYSTALLINE VACANCIES AND TO DISLOCATIONS MOVEMENT; BOTH PHENOMENA FOLLOW LAWS THAT REQUIRE THE OVERCOMING OF AN ENERGY THRESHOLD AND THAT CAN BE DESCRIBED BY THE ARRHENIUS LAW:

$$\dot{\varepsilon} = \frac{d\varepsilon}{dt} = A e^{-\frac{Q}{RT}}$$

WITH:

Q: ACTIVATION ENERGY

T: ABSOLUTE TEMPERATURE

R: UNIVERSAL GAS CONSTANT

A: MATERIAL CHARACTERISTIC (INDEPENDENT OF STRESS AND TEMPERATURE)

- INTEGRATING THE ARRHENIUS LAW IT FOLLOWS:

$$\varepsilon = A e^{-\frac{Q}{RT}} \cdot t$$

3
- OXIDATION REACTIONS ARE BASED ON DIFFUSION PHENOMENA THAT CAN BE DESCRIBED USING THE FICK LAWS:

– THE FIRST FICK LAW INTRODUCES THE DIFFUSION COEFF. (D) WHICH IS THE FLOW RATE PER UNIT AREA AND PER UNIT CONCENTRATION GRADIENT ON A DEFINED REFERENCE PLANE:

$$J = -D \frac{\partial C}{\partial x} \quad , \quad D = D_0 e^{-\frac{Q}{RT}}$$

WITH:

J: INSTANTANEOUS FLOW RATE PER UNIT AREA OF THE DIFFUSING SPECIES ACROSS A PLANE

C: CONCENTRATION OF THE SPECIES WHICH DIFFUSE ON THE REFERENCE PLANE

$\partial C / \partial x$: CONCENTRATION GRADIENT NORMAL TO THE PLANE

X: GENERIC SPATIAL POSITION

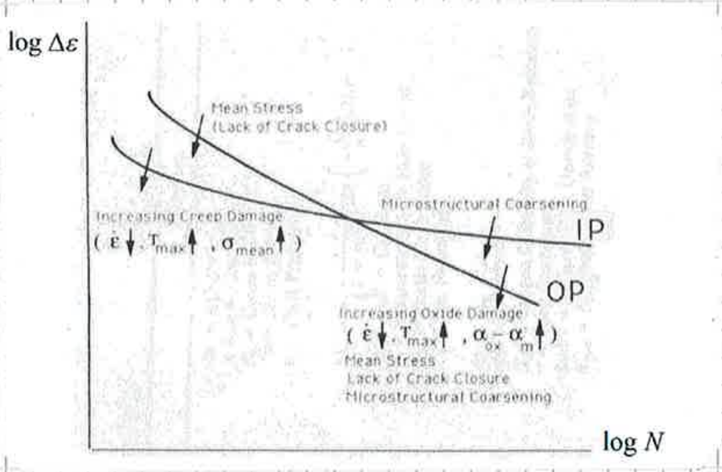
Q: ACTIVATION ENERGY

– THE SECOND FICK LAW DESCRIBES THE EVOLUTION OF C AS:

$$\dot{C} = \frac{dC}{dt} = \frac{\partial}{\partial x}\left(D \frac{\partial C}{\partial x}\right)$$

• TMF RESIDUAL LIFE ESTIMATION

• IN GENERAL, THE **COMPONENT LIFE** (IN TERMS OF NUMBER OF CYCLES TO FAIL.) **DEPENDS ON MATERIAL** AND **DAMAGE MECHANISM** AND CAN BE **ESTIMATED BY** MEANS OF $\Delta\varepsilon$-N CURVES OR CORRESPONDING MATHEMATICAL **MODELS**.



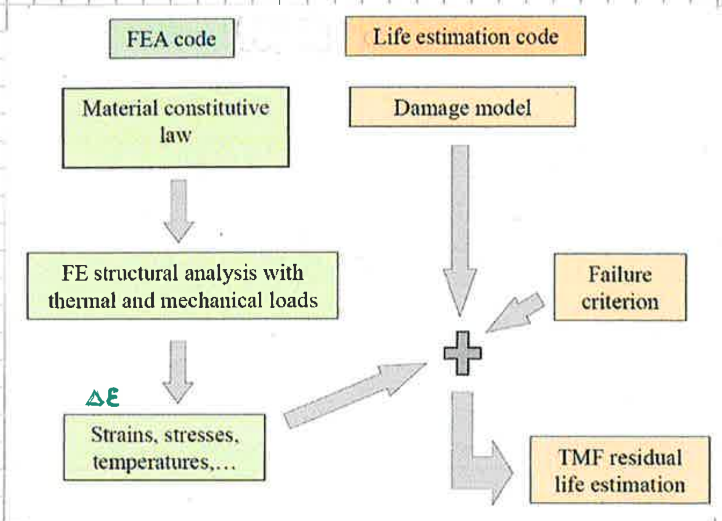
• **LITERATURE MODELS** TO PREDICT THE RESIDUAL LIFE OF MATERIAL SPECIMENS UNDER TMF CONDITION **ARE MANY**, BUT THEIR USE IN INDUSTRIAL FIELD IS STILL LIMITED.

• **THESE MODELS EVALUATE** THE **STRESS STATE** (DEPENDING ON THE DIFFERENT DAMAGE CAUSES FOR THE SPECIFIC MATERIAL AND THE SPECIFIC APPLICATION IN EXAM) AND **CALCULATE** AN **ESTIMATION OF THE NUMBER OF CYCLES TO FAILURE** (N_f).

• **ANY MODEL** IS **CONSTITUTED** BY **THREE MAIN PARTS**:

1. THE **MATERIAL CONSTITUTIVE LAW** WHICH CORRELATES STRESS AND STRAIN WITH DIFFERENT MATHEMATICAL RELATIONS (ELASTO-PLASTIC, VISCO-ELASTIC, VISCO-ELASTO-PLASTIC, PLASTIC WITH STRENGTHENING, ...) [illegible]
2. THE **DAMAGE MODEL** WHICH **DESCRIBES** WHAT ARE THE PHENOMENA THAT CAUSE THE MATERIAL DAMAGE (MECHANICAL FATIGUE, CREEP, OXIDATION,..) AS THEY ACT AND **HOW THEY EVOLVE** (E.G. DEPENDING ON STRESS, STRAIN, TEMPERATURE, ETC); THE DAMAGE CAN BE EXPRESSED BOTH IN FULL FORM AND DIFFERENTIAL FORM). [illegible]
3. THE **FAILURE CRITERION** WHICH **EXPRESSES** THE **LIMIT VALUE** FOR THE DAMAGE EXPRESSION: IF THE THRESHOLD IS EXCEEDED THE COMPONENT BREACKS. [illegible]

- TOGETHER WITH THE THERMO-MECHANICAL RESULTS APPROPRIATELY REORGANIZED, FAST-LIFE REQUIRES IN INPUT ALL THE VARIABLES FOR THE EVALUATION OF THE RESIDUAL LIFE, I.E. ALL THE PARAMETERS OF THE SELECTED DAMAGE MODEL)
- BY RUNNING THE ANALYSIS (UNIAXIAL OR MULTIAXIAL DEPENDING ON THE SELECTED DAMAGE MODEL) ON SPECIFIC COMPONENT AREAS (WHERE FEA HAS COMPUTED MAXIMUM PLASTIC DEFORMATIONS), FAST-LIFE CALCULATES THE RESIDUAL LIFE ASSOCIATED TO THE MOST CRITICAL FINITE ELEMENT OF EACH AREA OF INTEREST.



FEA CODE: ABACUS / ANSYS / NASTRAN (COMMERCIAL CODES)
LIFE ESTIMATION CODE: FE-SAFE / FEM-FAT (COMMERCIAL CODES) / FAST-LIFE

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- THE VISCOUS BEHAVIOUR TAKES INTO ACCOUNT THE TIME DEPENDENCE OF THE $\sigma - \varepsilon$ RELATIONSHIP THAT IS MANIFESTED IN TWO WAYS:
  FOR CONSTANT STRESSES THE STRAIN VARIES IN TIME OR VICE VERSA.
- THIS MEANS THAT IN STRAIN-CONTROLLED CYCLIC TESTS THE STRESS VALUES DEPEND ON THE STRAIN RATE, AND FOR STRESS-CONTROLLED CYCLIC TESTS THE STRAIN VALUES DEPEND ON THE STRESS RATE.
- THIS KIND OF DEPENDENCE CAN BE DESCRIBED BY MODELS MUCH MORE COMPLEX (E.G. MAXWELL-VOIGHT MODEL).

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- CHABOCHE, BMW (EMPIRICAL, STRAIN-RANGE PARTITIONING-TYPE) - DAMAGE TO MECHANICAL FATIGUE AND CREEP ARE TAKEN INTO ACCOUNT, THE FAILURE CRITERION IS BASED ON CUMULATIVE DAMAGE ASSUMPTION.
- SKELTON, ~~MODIFIED J-INTEGRAL, DANG VAN~~ (EMPIRICAL, ENERGY-TYPE) - DAMAGE IS COMPUTED AS A FUNCTION OF THE ENERGY DISSIPATED IN THE MATERIAL HYSTERESIS LOOPS, THE FAILURE CRITERION IS BASED ON A LIMIT VALUE OF DISSIPATED ENERGY.

• MULTIAXIAL DAMAGE MODELS - USED IN LCF OR TMF ANALYSIS - INVESTIGATE COMPONENTS INSTEAD OF SPECIMENS.

• SEVERAL MULTIAXIAL DAMAGE MODELS (VON MISES[1], ASME CODE[2], SONSINO-GRUBISIC[3], MANDIL-BROWN-MILLER[4], FATEMI-SOCIE[5]) ARE BASED ON THE BASQUIN-MANSON-COFFIN RELATION. (BMC MODEL)

• THEY CALCULATE AN EQUIVALENT UNIAXIAL STRAIN (OR SHEAR)

• THE COMPUTED UNIAXIAL EQUIVALENT STRAIN IS INTRODUCED IN BMC MODEL, OR IN ANOTHER UNIAXIAL MODEL, TO ESTIMATE THE COMPONENT NUMBER OF CYCLES TO FAILURE.

• (BASIC ASSUMPTIONS AND MATHEMATICAL FORMULATION OF THE EQUIVALENT UNIAXIAL STRAIN (OR SHEAR) ARE DIFFERENT FOR EACH MULTIAX. DAMAGE MODEL)

• (ALL MULTIAXIAL DAMAGE MODELS ARE CALIBRATED WITHOUT FURTHER EXPERIM. TESTS THAN THOSE NEEDED FOR THE CALIBRATION OF THE CHOSEN UNIAXIAL MODEL) (I.E. 4 PARAMETERS FOR BMC MODEL CALIBRATION)

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MECHANICAL FATIGUE

- HCF → $\Delta\sigma^k \cdot N = B$ (BASQUIN)
- LCF → STRAIN CONTR. → BETTER CHARACT. M. BEHAV. → $\Delta\varepsilon^{\beta} N = C$ (MANSON-C.)

THERMAL FATIGUE

- T CYCLE → DECR. OF MECH., PHYS. PROP.

TMF

- TH. AND MECH. CYCLES ARE SUPERIMPOSED IP OR OP

• MAT. PROP.
• DEF. RATE (FACTORS TO BE TAKEN INTO CONSID.)
• T RANGE
• PHASE CONDITION

EFFECT OF T : ΔT → STRAIN COMP. + ACT. OF CREEP AND OXIDATION

(CREEP = PROGRESSIVE STRAIN ACCUM. UNDER CONST. LOAD WHEN $T > \frac{1}{3} T_M$
OXIDATION = WHEN ENV. IS CHARACT. BY AGGRESSIVE GAS
EXHAUST MANIF. SUBJECTED TO BOTH)

→ PREDICTION OF MAT. BEHAVIOUR
→ RESEARCH OF THE CAUSE OF COMP. FAILURE (GOALS)
→ ESTIMATION OF N

• ISOTHERM. MECH. FATIGUE TEST : T = CONST LOAD = VARIABLE
• THERMAL FATIGUE TEST : T = VARIABLE LOAD = CONST (TESTS)
• THERMO-MECH. FATIGUE TEST : T = VARIABLE LOAD = VARIABLE

- VIBROFORE (DEVICES)
- SERVO-HYDRAULIC MACHINE

• STRAIN-CONTROLLED (IMPOSITION OF $\Delta\varepsilon_{TOT}$) (CONTROL MODES)
• STRESS-CONTROLLED (IMPOSITION OF $\Delta\sigma$, $\sigma_m \neq 0$)

CRACKS

- FATIGUE C. STARTS FROM EXT. SURF. OF COMP.
- CREEP C. ARE INTERNAL; BIGGER THAN OX. C.
- OXIDATION C. ARE INTERNAL, INTER-GRANULAR

UNIAXIAL DAMAGE MODEL

- BASQUIN - MANSON - COFFIN : SIMPLEST, MOST USED IN LCF (ALSO IN TMF)

- ONLY MECHANICAL FATIGUE (NO CREEP, NO OXIDATION)
- SUITABLE FOR "NOT HIGH T" "NOT LONG WORKING"

MATERIAL CONST LAW = ELASTO - PLASTIC HARDENING TYPE (1)

$$\Delta\varepsilon = \Delta\varepsilon_{EL} + \Delta\varepsilon_{PL} = \frac{\Delta\sigma}{E} + \left(\frac{\Delta\sigma}{K'}\right)^{1/m'}$$

MODEL: (2)

$$\Delta\varepsilon = \Delta\varepsilon_{EL} + \Delta\varepsilon_{PL} = \frac{\sigma_f'}{E} N_f^b + \varepsilon_f' N_f^c$$

(CALIBRATION σ_f', b, ε_f', c -> EXPERM. DATA FROM HCF, LCF TESTS)

FAILURE CRITERION (3)

$$FEA \rightarrow \Delta\varepsilon = \frac{\sigma_f'}{E} N_f^b + \varepsilon_f' N_f^c \rightarrow N_f$$

- MORROW

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- UDM - NEU-SEHITOGLU MODEL

- NEU-SEHITOGLU MODEL SUITABLE FOR LOAD HISTORIES WITH SUPERIMPOSED MECHANICAL AND THERMAL CYCLES.
- IT IS THEN NATURALLY SUITABLE FOR TMF APPLICATION BUT REQUIRES THE EXPERIMENTAL CALIBRATION OF MANY PARAMETERS WITH RESPECT THE BASQUIN-MANSON-COFFIN MODEL.
- THE CONSTITUTIVE LAW IS OF VISCO-PLASTIC (WITH CYCLIC HARDENING) TYPE
- THE TOTAL DAMAGE, EQUAL TO THE INVERSE OF THE NUMBER OF CYCLES TO FAILURE, IS DEFINED AS THE SUM OF THE CONTRIBUTIONS DUE TO MECHANICAL FATIGUE (IN PLASTIC RANGE, WITH STRESSES HIGHER THAN YIELD STRENGTH), CREEP (PRODUCED BY EQUIVALENT STRESSES LOWER THAN YIELD STRENGTH AND HIGH TEMPERATURE) AND OXIDATION:

$$D = D^{FAT} + D^{CREEP} + D^{OX} \Rightarrow \frac{1}{N_f} = \frac{1}{N_f^{FAT}} + \frac{1}{N_f^{CREEP}} + \frac{1}{N_f^{OX}}$$

- EACH CONTRIBUTION TO THE TOTAL DAMAGE IS INDEPENDENTLY CALCULATED.
- THE AMOUNT OF MECHANICAL FATIGUE DAMAGE IS CALCULATED BY THE MODEL BASQUIN-MANSON-COFFIN:

$$D^{FAT} = \frac{1}{N_f^{FAT}}, \text{ WITH } N_f^{FAT} \text{ FROM } \Delta\varepsilon_{TOT} = \frac{\sigma_f'}{E} N_f^{b} + \varepsilon_f' N_f^{c} \quad \text{(4 PARAMETERS)}$$

- THE AMOUNTS OF CREEP AND OXIDATION DAMAGE REQUIRE EXTENSIVE AND EXPENSIVE TESTING FOR CALIBRATION OF THE MANY PARAMETERS PRESENT IN THEIR EXPRESSIONS: (+ 14 PARAMETERS: 7 FOR CREEP; 7 FOR OXIDATION)

$$D^{CREEP} = \Phi^{CREEP} \int_0^t A e^{(-\Delta H/RT)} \left(\frac{\alpha_1 \sigma_{EQ} - \alpha_2 \sigma_H}{K} \right)^m dt$$

$$D^{OX} = \left[\frac{h_{cr} \delta_0}{B \Phi^{ox} K_p^{EFF}} \right]^{-\frac{1}{\beta}} \cdot \frac{2 \Delta\varepsilon^{(1+2/\beta)}}{\dot{\varepsilon}^{(1-\alpha/\beta)}}$$

• ENERGY-TYPE DAMAGE MODELS

- ETDM - SKELTON MODEL

1 • SKELTON MODEL IS BASED ON AN **ENERGY APPROACH** WHICH CORRELATES THE ENERGY DISSIPATED IN TMF CONDITION TO THE CRACK PROPAGATION.

2 • THE COMPONENT **RESIDUAL LIFE** IS ESTIMATED DIVIDING THE **TOTAL ENERGY** THAT THE **MATERIAL** IS ABLE TO DISSIPATE UP TO FAILURE BY THE ENERGY DISSIPATED BY THE **COMPONENT** DURING ITS **WORKING CONDITION**.

3 • FROM THE ANALYSIS OF LCF EXPERIMENTAL TESTS ON DIFFERENT MATERIALS, SKELTON DEVELOPED HIS APPROACH IMPUTING THE **OVERALL ENERGY EXPENDITURE** TO THE **HEAT DISSIPATION** THAT COMES FROM THE SUCCESSION OF THE **HYSTERESIS CYCLES** (BOTH NON STABILIZED AND STABILIZED).

4 • THE MEASURE OF THE **DISSIPATED ENERGY** IS REPRESENTED BY THE **TOTAL AREA OF THE HYSTERESIS CYCLES UP TO FAILURE**.

5 • BECAUSE SKELTON MODEL IS AN **ENERGY TYPE MODEL**, IT NATURALLY TAKES INTO ACCOUNT THE ACTUAL STRESS STATE WITHOUT DISTINGUISHING BETWEEN SINGLE AND MULTIAXIAL CONDITION.

6 • THE **PLASTIC WORK PER UNIT VOLUME** OF MATERIAL, CORRESPONDING TO THE **AREA OF THE i-th HYSTERESIS LOOP**, CAN BE WRITTEN AS:

$$W_i = \frac{1}{1+\beta}\left(C\Delta\varepsilon N_i^{\lambda} - \frac{C^2 N_i^{2\lambda}}{E}\right)$$

(NOT SEEN DURING 2020-2021 E. LESSON)

WITH:

β: CYCLIC HARDENING PARAMETER

λ: EVOLUTIONARY EXPONENT

C: RESISTANCE PARAMETER OF THE MATERIAL

7 • SUMMING THE PLASTIC WORK SPENT IN EACH CYCLE (E.G. AFTER 10, 50, 200, 1000 C.) AND REPLACING THE SUMMATION WITH THE INTEGRAL (FOR A SUFFICIENTLY HIGH n. OF C.) THE **TOTAL ENERGY EXPENDITURE** AFTER N CYCLES CAN BE CALCULATED:

$$W = \sum_{i=1}^{N} w_i = \frac{1}{1+\beta}\left(\frac{C\Delta\varepsilon N^{1+\lambda}}{1+\lambda} - \frac{C^2 N^{1+2\lambda}}{E(1+2\lambda)}\right)$$

SKELTON MODEL TAKES INTO ACCOUNT THE **ACTUAL STRESS** (UNIAXIAL, MULTIAXIAL) AND THE **ACTUAL DAMAGE SOURCE** (FATIGUE, CREEP, OXIDATION)

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• MULTIAXIAL DAMAGE MODELS

- MULTIAXIAL DAMAGE MODELS ARE DEVELOPED TO BE USED IN LCF AND TMF ANALYSIS OF COMPONENTS WHERE THE STRESS STATE IS GENERALLY MULTIAXIAL.

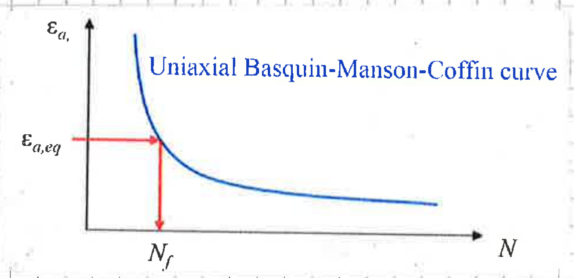
1 • MOST OF THE MULTIAXIAL DAMAGE MODELS ARE BASED ON THE BASQUIN-MANSON-COFFIN MODEL, EVENTUALLY CORRECTED (MORROW OR SMITH-WATSON-TOPPER CRITERIA) IN CASE OF NON-ZERO MEAN STRESS.

2 • ALL MULTIAXIAL DAMAGE MODELS CALCULATE A UNIAXIAL STRAIN AMPLITUDE EQUIVALENT TO THE MULTIAXIAL CONDITION ACTING ON THE COMPONENT. THIS EQUIVALENT UNIAXIAL STRAIN AMPLITUDE IS THEN INTRODUCED IN THE BASQUIN-MANSON-COFFIN MODEL (EXPRESSED IN TERMS OF ε_a INSTEAD OF $\Delta\varepsilon$) TO ESTIMATE THE NUMBER OF CYCLES TO FAILURE, I.E. THE COMPONENT RESIDUAL LIFE:

EXPERIMENTALLY CALIBRATED

$$\varepsilon_{a,eq} = \frac{\sigma_f'}{E}(2N)^b + \varepsilon_f'(2N)^c \Rightarrow N_f \quad \text{(RESIDUAL LIFE ESTIMATION)}$$

3 • DEPENDING ON THE CONSIDERED MULTIAXIAL DAMAGE MODEL, THE CALCULATION OF THE EQUIVALENT UNIAXIAL STRAIN IS BASED ON DIFFERENT ASSUMPTIONS AND HAS A DIFFERENT FORMULATION.



4 • MULTIAXIAL DAMAGE MODELS ARE CLASSIFIED IN STRAIN-BASED MODELS AND MODELS BASED ON THE CRITICAL PLANE.

5 • THE FIRST CATEGORY INCLUDES THE VON MISES, ASME CODE AND SONSINO-GRUBISIC MODELS, THE SECOND CATEGORY THE KANDIL-BROWN-MILLER AND THE FATEMI-SOCIE MODELS.

6 • ALL THESE MODELS ARE FORMULATED YET SIMPLE ENOUGH AND ARE AUTOMATICALLY CALIBRATED WITHOUT FURTHER TESTS THAN THOSE ARE NEEDED FOR THE CALIBRATION OF THE PARAMETERS OF THE BASQUIN-MANSON-COFFIN RELATIONSHIP.

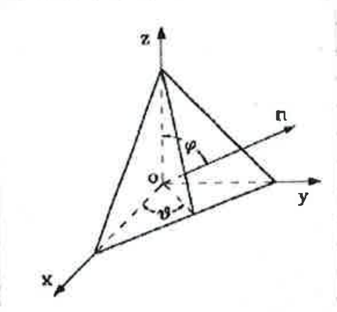
FROM COMPONENT FEA → MULTIAXIAL STRESS AND STRAIN STATES → MDM (3D) → $\varepsilon_{a,eq}$ → BMC (1D) → N_f

/NEU-SIET
/CHABOCHE
/SKELTON

- MDM - SONSINO - GRUBISIC MODEL

- THE SONSINO-GRUBISIC MODEL ASSUMES THAT THE DAMAGE AND THE CONSEQUENT COMPONENT LIFE REDUCTION UNDER CONDITIONS OF OP TMF IS CAUSED BY THE CHANGE OF THE MAIN DIRECTIONS OF STRAIN DUE TO THE INTERACTION OF THE DEFORMATIONS ON THE COMPONENT SURFACE.
- THIS INTERACTION IS TAKEN INTO ACCOUNT CONSIDERING THE ARITHMETIC MEAN OF THE SHEAR STRAIN AMPLITUDES ACTING ON AN ELEMENTARY SUPERFICIAL PLANE, NAMED INTERFERENCE PLANE AND DEFINED BY THE UNIT VECTOR ($\vec{n}$) PERPENDICULAR TO THE SURFACE, AND USING THE SPHERICAL COORDINATES θ, φ.
- FOR SIMPLICITY, THE INTERFERENCE PLANE IS ASSUMED PERPENDICULAR TO THE COMPONENT SURFACE (THUS φ COORDINATE IS CONSTANT AND EQUAL TO 90°)
- THE SHEAR STRAIN IS CALCULATED AS:

$$\gamma(\theta,t) = [\varepsilon_{yy}(t) - \varepsilon_{xx}(t)]\sin(2\theta) + \varepsilon_{xy}(t)\cos(2\theta)$$



- THE CALCULATION IS PERFORMED BY DISCRETIZING THE ANGLE θ AND, FOR EACH ANGLE θ_i, THE MAXIMUM AND THE MINIMUM VALUES USEFUL TO CALCULATE THE AMPLITUDE OF THE SHEAR STRAIN ARE COMPUTED.
- THE SHEAR STRAIN AMPLITUDE IS THEN COMPUTED AS:

$$\gamma_a(\theta_i) = \frac{MAX[\gamma(\theta_i,t)] - min[\gamma(\theta_i,t)]}{2}$$

- BY INTEGRATING THE $\gamma_a(\theta_i)$ ON THE INTERFERENCE PLANE AND DIVIDING BY π THE ARITHMETIC MEAN OF THE SHEAR STRAIN AMPLITUDES IS OBTAINED AND THE UNIAXIAL EQUIVALENT STRAIN AMPLITUDE COMPUTED:

$$\varepsilon_{a,EQ} = \frac{5}{9(1+\nu)} \gamma_{a,ARITHM.}$$

VON MISES AND ASME CODE MODELS ARE THE MOST GENERAL MULTIAXIAL DAMAGE MODEL.
=> FOR ANY APPLICATION OF TMF.

SONSINO - GRUBISIC MODEL IS WELL APPLIED WHEN WE WANT TO INVESTIGATE IN PARTICULAR THE OP TMF: THE OP TMF CONDITIONS PRODUCE VARIATIONS IN TIME OF THE PRINCIPAL STRAIN COORDINATES. (BETTER THAN IP TMF)

KANDL-BROWN-MILLER MODEL IS SUITABLE TO BE APPLIED FOR APPLICATIONS THAT INVOLVE LARGE PLASTIC DEFORMATIONS.

FATEMI-SOCIE MODEL IS SUITABLE FOR APPLICATIONS IN WHICH SHEAR IS THE DOMINANT STRESS.

- THE **MAXIMUM SHEAR STRAIN** IS CALCULATED AS IN KANDIL-BROWN-MILLER MODEL ($\Delta\gamma_{MAX} = \varepsilon_1 - \varepsilon_3$) AND THE **MAXIMUM NORMAL STRESS** IS CALCULATED AS THE MEAN VALUE BETWEEN THE MAXIMUM AND THE MINIMUM PRINCIPAL STRESSES ($\sigma_{m,MAX} = (\sigma_1 + \sigma_3)/2$)
- **SUBSTITUTING** IN THE PREVIOUS FORMULATION THE FATEMI-SOCIE MODEL CAN BE REWRITTEN AS:

$$\gamma_{aEQ} = \frac{\varepsilon_1 - \varepsilon_3}{2}\left(1 + K_{FS}\frac{\sigma_1 + \sigma_3}{2\sigma_y}\right)$$

- **TO ESTIMATE** THE **RESIDUAL LIFE** THE **BASQUIN-MANSON-COFFIN LAW** MUST BE OBVIOUSLY **REWRITTEN** IN TERMS OF **SHEAR STRAIN**:

$$\gamma_{aEQ} = \frac{\tau_f'}{E}(N_f)^{b_\gamma} + \gamma_f'(N_f)^{c_\gamma} \Rightarrow N_f$$

WITH:

$$\tau_f' \approx \frac{\sigma_f'}{\sqrt{3}} \; ; \; \gamma_f' \approx \frac{\varepsilon_f'}{\sqrt{3}} \; ; \; b_\gamma \approx b \; ; \; c_\gamma \approx c$$

$\varepsilon_1, \varepsilon_3$ MAX. AND MIN. PRINCIPAL STRAINS;

σ_1, σ_3 MAX. AND MIN. PRINCIPAL STRESSES.

- GEOMETRY AND MATERIALS

- IN MODERN DIESEL ENGINES THE COMMON GEOMETRY IS THAT OF TURBOCHARGED ENGINES : SHORT RUNNERS WITH A UNIQUE OUTLET SECTION TO GUIDE EXHAUST GASSES DIRECTLY TO THE TURBOCHARGER.
- THE MOST USED MATERIALS ARE:

- STEEL (SINGLE SHEET TO REDUCE WEIGHT OR DOUBLE-SHEET TO REDUCE HEAT DISSIPATION) : AISI 409, AISI 441, AISI 429, AISI 444, AISI 321. (AISI 400 FAMILY)
- Si-Mo OR Si-Mo-Cr DUCTILE CAST IRON.
- Ni-RESIST DUCTILE CAST IRON (FOR VERY HIGH TEMPERATURE CONDITIONS)

Cast iron exhaust manifold (FIAT SDE 1.3 16v)

Cast iron exhaust manifold with turbocharger (VW 2.0 TDI)

Sheet steel exhaust manifold (Honda 2.2 i-CTDi)

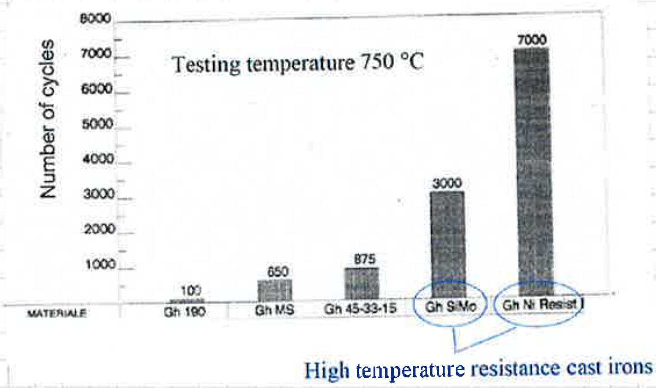
IN CASE OF USE OF STEEL THE E.M. PRESENTS A WELDED ZONE THAT CONNECTS THE UPPER PART OF THE MAIN COLLECTOR TO THE LOWER PART OF THE MAIN COLLECTOR.

STEEL USE IS USUALLY LIMITED TO INDUSTRIAL APPLICATIONS AND SOME GASOLINE PASSENGER E. APPLICATIONS (← CAST IRON IS PREFERRED FOR THE DUCTILE CAPABILITY AT HIGH T.)

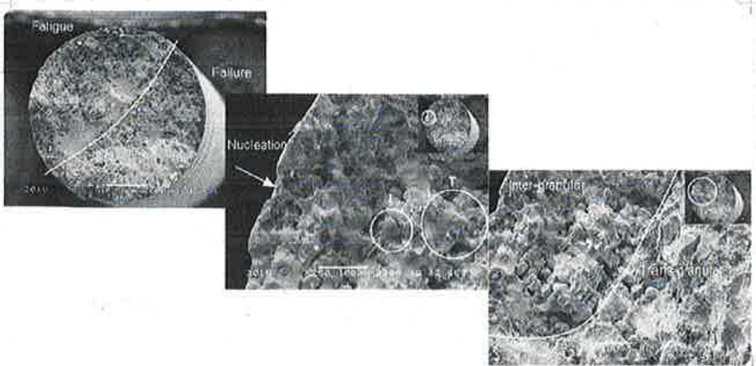
- THERMAL SHOCK TEST:
  - ENGINE IS SUBJECTED TO FAST SPEED REGIME VARIATIONS (FROM MINIMUM TO MAXIMUM SPIN SPEED IN FEW SECONDS)
  - COOLING PHASE IS DUE TO A COLD AIR FLUX THAT ACTS DIRECTLY ON THE EXHAUST MANIFOLD.
  - THESE EXTREME WORKING CONDITIONS ARE IMPOSED TO OBTAIN A FATIGUE LIFE WITHIN 5000 CYCLES DURING THE TEST. (IN FOLLOWING CASE-STUDY $N_S = 3400$ [CYCLES])
- OTHER INVESTIGATED TESTS ARE:
  - FUJI FILM TEST: A FILM SENSITIVE TO PRESSURE IS PLACED BETWEEN EXHAUST MANIFOLD AND CYLINDER HEAD GASKET AND BETWEEN CYLINDER HEAD GASKET AND CYLINDER HEAD. VARYING BOLTS TORQUE AND MANIFOLD HEATING, THE FILM IMPRESSION IS ANALYZED TO VERIFY POTENTIAL GAS LEAKAGE.
  - DIRECT GAS LEAKAGE TEST: TO REVEAL GAS LEAKAGES DUE TO FATIGUE CRACKS.

- MATERIAL BEHAVIOUR

- CAST IRON FAILURE WITH RESPECT TO ISOTHERMAL HIGH TEMPERATURE LCF DEPENDS ON MATERIAL TEMPERATURE RESISTANCE.



- FAILURE OF HIGH T RESISTANCE CAST IRON (Si-Mo, Si-Mo-Cr AND Ni-RESIST) IS THE COMBINATION OF INTER-GRANULAR AND TRANS-GRANULAR CRACK PROPAGATION.



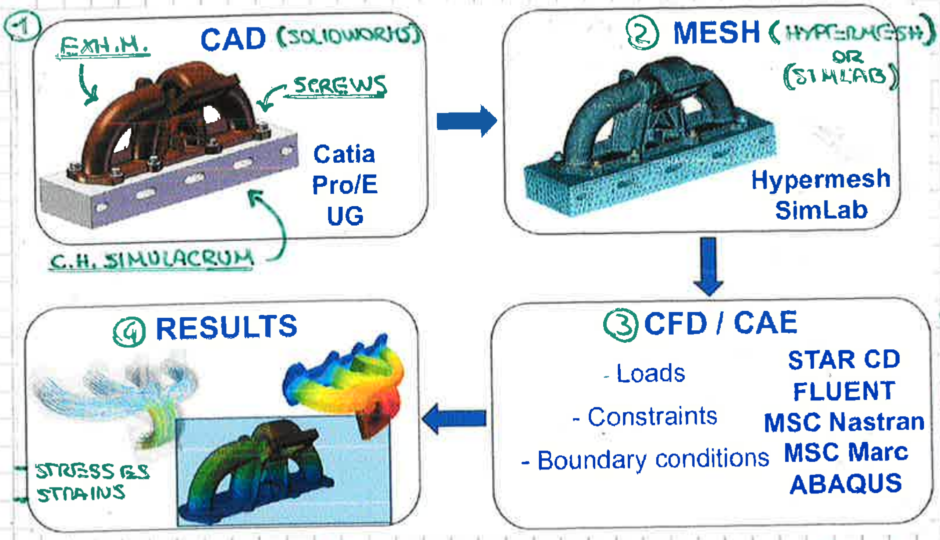
- BASQUIN-MANSON-COFFIN MODEL PARAMETERS AND SKELTON MODEL PARAMETERS OF A COMMERCIAL Si-Mo-Cr CAST IRON FOR EXHAUST MANIFOLD APPLICATION ARE AVAILABLE FOR DIFFERENT TEMPERATURE VALUES (PROC. OF ASME 2014 12th BIENNAL CONFERENCE ON ENGINEERING SYSTEMS DESIGN AND ANALYSIS - ESDA 2014-20458)

BMC: σ'_f, b, ε'_f, c — SKELTON: W_c

T [°C]	σ'_f [MPa]	b	ε'_f	c	W_c [J/mm^3]
160	2857	–0.2205	0.1221	–0.7199	0.644
500	1596	–0.2127	0.0721	–0.4619	1.785
800	585	–0.1892	0.0349	–0.5878	0.057

- SINCE THERE ISN'T A FULL SCIENTIFIC AGREEMENT ABOUT THE INFLUENCE OF CREEP, IN MOST NUMERICAL INVESTIGATIONS CREEP PHENOMENON IS NEGLECTED WHILE THE PRESENCE OF OXIDATION RECOMMENDS CHOOSING A DAMAGE MODEL ABLE TO TAKE INTO ACCOUNT THE OXIDATION DAMAGE MECHANISM.

- RLE - PROCEDURE



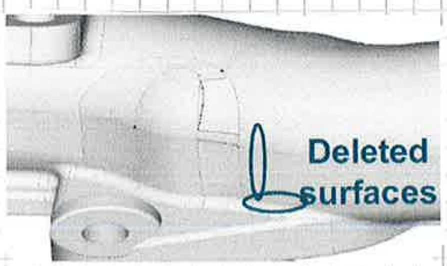
③ RLE { MAT. CONST. LAW. / FEA / RLE - MULTI-AX. D. MODELS

DIFFERENT CODES:
- TO SIMULATE THE PRESENCE OF EXHAUST GASES
  → STAR CD / FLUENT
- TO SIMULATE MECH LOADS AND CONSTRAINTS
  → NASTRAN, MARC, ABACUS (FE CODES) OR ANSIS

⑤ IDENTIFICATION OF CRITICAL AREAS IN TERMS OF PLASTIC DEFORMATION

- RLE - CAD

- SEVERAL FUNDAMENTAL STEPS ARE INVOLVED IN THE MODEL PREPARATION:
  - GEOMETRY AND MESH IMPORT FROM APPROPRIATE CODES;
  - LOADS AND CONSTRAINTS DEFINITION ON SEVERAL AREAS OF THE MODEL: CONCENTRATED FORCES, PRESSURES, INERTIAS, DISTRIBUTED LOADS, DISPLACEMENTS, VELOCITIES, ACCELERATIONS, ...
  - MATERIAL DEFINITION SPECIFYING ALL THE REQUIRED PARAMETERS: MASS DENSITY, ELASTICITY AND PLASTICITY PROPERTIES, THERMAL EXCHANGE PARAMETERS, ...
  - PROPERTIES DEFINITION OF THE MODEL: ASSIGNMENT OF MATERIAL PROP. TO THE MODEL ELEMENTS.
- THE GEOMETRIES RESULTING FROM CAD ARE USUALLY NOT OPTIMIZED AND TOO SMALL OR DISTORTED SURFACES CAN BE PRESENT; IT IS POSSIBLE TO AUTOMATICALLY CORRECT WITH THE "GEOMETRY CLEANUP" COMMAND.



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- RLE - MATERIAL CONSTITUTIVE LAW

- MATERIAL CONSTITUTIVE LAW IS STRICTLY DEPENDENT ON THE SPECIFIC APPLICATION AND A GOOD COMPROMISE FOR THERMO-STRUCTURAL ANALYSIS OF Si-Mo, Si-Mo-Cr AND Ni-RESIST CAST IRON EXHAUST MANIFOLD IS THE SO-CALLED COMBINED ISOTROPIC-KINEMATIC HARDENING MODEL.
- MODEL PARAMETERS CAN BE ESTIMATED BY STATIC AND (≠) ISOTHERMAL LCF EXPERIMENTAL TESTS AT DIFFERENT TEMPERATURES (RT, 200[°C], 500[°C], 800[°C] FOR Si-Mo AND Si-Mo-Cr CAST IRON, ALSO 1000[°C] FOR Ni-RESIST CAST IRON). (RT: ROOM TEMPERATURE)
- MODEL PARAMETERS AT OTHER TEMPERATURE LEVELS CAN BE ESTIMATED BY INTERPOLATING THE PREVIOUS EXPERIMENTAL RESULTS.
- A POSSIBLE EXAMPLE OF EXPERIMENTAL TEST CAMPAIGN ON Si-Mo AND Si-Mo-Cr CAST IRON IS:
  - STATIC TESTS AT 4 TEMPERATURE LEVELS (RT, 160[°C], 500[°C] AND 800[°C]), AT LEAST 3 SPECIMENS FOR EACH LEVEL.
  - SYMMETRIC LCF TESTS AT 4 TEMPERATURE LEVELS (RT, 160[°C], 500[°C], AND 800[°C]), 3 APPLIED STRAIN RANGES FOR EACH LEVEL (0.3%, 0.4%, 0.5%), AT LEAST 3 SPECIMENS FOR EACH CONDITION.

- RLE - FEA

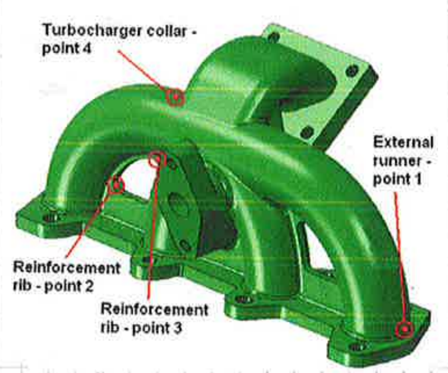
- TWO FE SIMULATIONS HAVE BEEN PERFORMED:

1) THERMAL TRANSIENT ANALYSIS:
   - TEMPERATURES ACHIEVED DURING THE IMPOSED THERMAL CYCLES ARE COMPUTED; TEMPERATURE RESULTS ARE COMPARED WITH THE EXPERIMENTAL VALUES MEASURED ON TEST BENCH.
   - NODAL TEMPERATURES COMPUTING BY THIS FIRST FEA ARE THE INPUT VALUES FOR THE SUBSEQUENT STRUCTURAL ANALYSIS.
2) STRUCTURAL TRANSIENT ANALYSIS:
   - STRESSES AND STRAINS DUE TO MECHANICAL LOADS (BOLTS AND TURBOCHARGER WEIGHT) ARE FIRSTLY COMPUTED.
   - THEN STRESSES AND STRAINS DUE TO THE THERMAL EXPANSION, ACCORDING TO THE TEMPERATURE MAPS COMPUTED IN THE THERMAL TRANSIENT ANALYSIS AND IMPOSED ON THE FE MODEL, ARE COMPUTED AND SUPERIMPOSED TO THE MECHANICAL PREVIOUS RESULTS.

- RLE - MULTIAXIAL DAMAGE MODELS

- RESIDUAL LIFE WAS ESTIMATED USING DIFFERENT MULTIAXIAL DAMAGE MODELS: VON MISES, ASME CODE, SONSINO-GRUBISIC, KANDIL-BROWN-MILLER, FATEMI-SOCIE, SHELTON.
- ALL THE DAMAGE MODELS WERE CALIBRATED THROUGH DATA OBTAINED BY EXPERIMENTAL TESTS CAMPAIGN IN TEMPERATURE RANGE BETWEEN 160 [°C] AND 800 [°C].
- SINCE ALL CONSIDERED MULTIAXIAL DAMAGE MODELS PREDICT THE RESIDUAL LIFE BY USING BASQUIN-MANSON-COFFIN LAW, IT IS SUFFICIENT TO CALIBRATE ITS FOUR PARAMETERS.
- AS THE BASQUIN-MANSON-COFFIN LAW DOES NOT TAKE INTO ACCOUNT ANY TEMPERATURE-DEPENDENT RELATION, THE CALIBRATION PROCESS IS PERFORMED FOR EACH TEMPERATURE LEVEL UNDER INVESTIGATION.
- LCF AND HCF EXPERIMENTAL RESULTS ARE USED TO OBTAIN THE MATERIAL PARAMETERS OF BASQUIN-MANSON-COFFIN AND OF SHELTON MODELS.

- RESIDUAL LIFE VALUES HAVE BEEN COMPUTED FOR EACH OF THE FOUR CRITICAL FE (POINTS 1,2,3,4) DETECTED IN STRUCTURAL FEA.



- AS FOR THE EXHAUST MANIFOLD UNDER INVESTIGATION NO EXPERIMENTAL FATIGUE LIFE DATA ARE AVAILABLE, TO VERIFY IF THE MOST CRITICAL FE DETECTED BY EACH DAMAGE CRITERIA IS CHARACTERIZED BY A REALISTIC FATIGUE LIFE, A REFERENCE EXPERIMENTAL RESIDUAL LIFE HAS BEEN FOUND IN LITERATURE FOR A VERY SIMILAR COMPONENT SUBJECTED TO ALMOST THE SAME LOADING COND.
- THIS REFERENCE RESIDUAL LIFE HAS BEEN EVALUATED IN 3400 [CYCLES]
- THE RESIDUAL LIFE ESTIMATIONS COMPUTED WITH THE DIFFERENT DAMAGE MODELS ARE COMPUTED EACH OTHER AND WITH THE REFERENCE VALUE (VISIBLE AS DASHED BLACK LINE IN FOLLOWING FIGURE).

E1 CRANK MECHANISM

FORMULARY

CONTENT

- THE CRANK MECHANISM TRANSFORMS THE THERMAL ENERGY OF THE COMBUSTION CHARGE INTO MECHANICAL ENERGY BY MEANS OF THE ENGINE WORKING CYCLE WHICH CONSISTS IN A SEQUENCE OF STROKES: INTAKE, COMPRESSION, EXPANSION, EXHAUST.
  INTAKE, EXPANSION STROKES: DOWNHILL MOTION
  COMPRESSION, EXHAUST: UPHILL MOTION
  4 STROKES ENGINE ≠ 2 STROKES ENGINE
- THE CRANK MECHANISM IS THE MECHANISM FOR WHICH WE ARE ABLE TO CONVERT THE RECIPROCATING MOTION OF THE PISTON INTO ROTATIONAL MOTION OF THE CRANKSHAFT.
- THE MAIN CHARACTERS OF CRANK MECHANISM ARE: PISTON, CONNECTING ROD, CRANK, C.B.
  THE PISTON IS CONNECTED TO THE CONNECTING ROD SMALL END THROUGH THE WRIST PIN;
  THE CONNECTING ROD BIG END IS CONNECTED TO THE CRANK THROUGH THE CRANK PIN;
- THE INVESTIGATION OF THE CRANK MECHANISM IS DEVELOPED IN TERMS OF:
  - KINEMATICS, IN ORDER TO CALCULATE DISPLACEMENTS, VELOCITIES AND ACCELERATIONS OF PISTON AND CONNECTING ROD;
  - DYNAMICS, IN ORDER TO EVALUE FORCES AND MOMENTS, ACTING IN THE C. MECHANISM, IN PARTICULAR ON PISTON, CONNECTING ROD, CRANK AND CYL. BLOCK.
- WE REDUCE THE SYSTEM TO A LUMPED PARAMETER MODEL: SIGNIFICANT GEOMETRICAL POINTS ARE THE POINTS REPRESENTING THE CRANKSHAFT ROTATION AXIS (O), THE POSITION OF THE PISTON (A) AND THE CRANK PIN ROTATION AXIS (B).
- WE DISTINGUISH BETWEEN TWO LAYOUT:
  - CENTERED LAYOUT: THE CYL. AXIS PASSES THROUGH THE WRIST PIN AXIS AND THE CRANKSHAFT ROTATION AXIS.
  - OFFSET LAYOUT: THE CYL. AXIS PASSES THROUGH THE WRIST PIN AXIS BUT, W.R.T. THE CENTERED LAYOUT, IS TRANSLATED IN AGREEMENT WITH THE ROTATION DIRECTION (WHEN APPROACHING THE TDC) BY A DISTANCE CALLED OFFSET. THIS RESULTS USEFUL:
    → TO REDUCE THE PRESSURE ACTING ON THE CONNECTING ROD SIDE;
    → TO REDUCE THE PISTON SLAP PHENOMENON.
- IN FACT, DURING DOWNHILL AND UPHILL MOTIONS THE CRANKSHAFT ROTATION ANGLE IS RESPECTIVELY SLIGHTLY GREATER AND LOWER THAN 180°. THIS IS TRANSLATED IN A LOWER INCLINATION OF THE CONNECTING ROD DURING DOWNHILL MOTION AND IN AN HIGHER INCLINATION DURING THE UPHILL MOTION.

SMALL END = FOOT
BIG END = HEAD

- K - PISTON DISPLACEMENT, OFFSET

$z_0 = z_{0,c} + z_{0,wp}$: OVERALL OFFSET

$DB = z_0 + \ell \sin\beta = R \sin\theta$

$\sin\beta = \frac{R}{\ell}\sin\theta - \frac{z_0}{\ell} = \Lambda \sin\theta - \delta$; $\cos\beta = \sqrt{1-(\Lambda\sin\theta-\delta)^2}$

$\Lambda = \frac{R}{\ell} = 0.20 \div 0.35$: ELONGATION RATIO

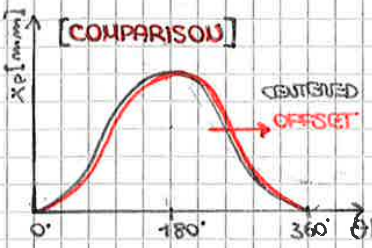
$\delta = \frac{z_0}{\ell} = 0 \div 0.045$: ADIMENSIONAL OFFSET

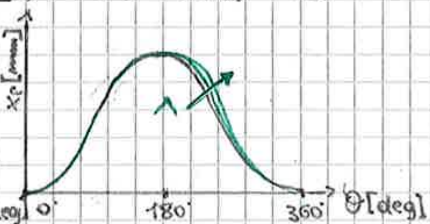
$OE = OA'\sin\varphi \Rightarrow \sin\varphi = \frac{OE}{OA'} = \frac{z_0}{R+\ell} = \frac{\delta}{(\Lambda+1)}$; $\cos\varphi = \sqrt{1-\left(\frac{\delta}{1+\Lambda}\right)^2}$

$$x_p = A'E - AE = A'E - AD' - D'E = (R+\ell)\cos\varphi - \ell\cos\beta - R\cos\theta =$$

$$= (R+\ell)\sqrt{1-\left(\frac{\delta}{1+\Lambda}\right)^2} - \ell\sqrt{1-(\Lambda\sin\theta-\delta)^2} - R\cos\theta =$$

$$= R\left[\left(1+\frac{1}{\Lambda}\right)\sqrt{1-\left(\frac{\delta}{1+\Lambda}\right)^2} - \frac{1}{\Lambda}\sqrt{1-(\Lambda\sin\theta-\delta)^2} - \cos\theta\right]$$: PISTON DISPLAC.



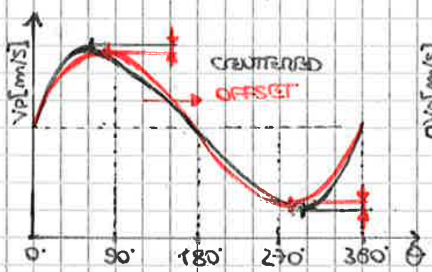


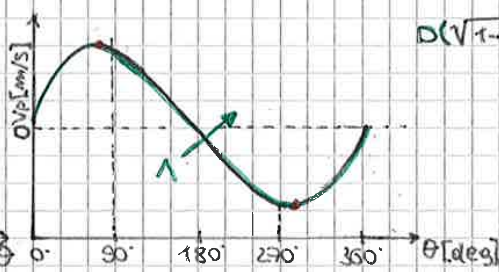
- K - PISTON VELOCITY, OFFSET

$$v_p = \frac{dx_p}{dt} = \frac{dx_p}{d\theta}\frac{d\theta}{dt} = \omega\frac{dx_p}{d\theta} = \omega R\left[\sin\theta - \frac{1}{\Lambda}\frac{1}{2}\frac{-2(\Lambda\sin\theta-\delta)\cdot\Lambda\cos\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}}\right] = \omega R\left[\sin\theta + \frac{\cos\theta(\Lambda\sin\theta-\delta)}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}}\right]$$

$$= \omega R\left[\sin\theta + \frac{\Lambda\sin\theta\cos\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}} - \frac{\delta\cos\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}}\right] = \omega R\left[\sin\theta + \frac{\Lambda}{2}\frac{\sin(2\theta)}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}} - \frac{\delta\cos\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}}\right]$$

$$D\left(\sqrt{1-(\Lambda\sin\theta-\delta)^2}\right) = \frac{1}{2}\frac{-2(\Lambda\sin\theta-\delta)\Lambda\cos\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}}$$



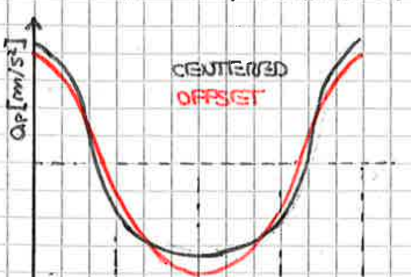


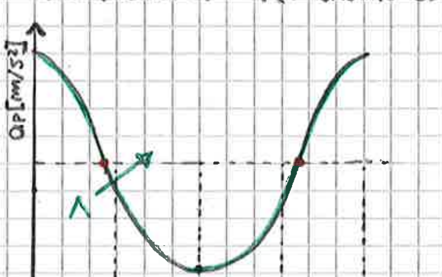
- K - PISTON ACCELERATION, OFFSET

$$a_p = \frac{dv_p}{dt} = \frac{dv_p}{d\theta}\frac{d\theta}{dt} = \omega\frac{dv_p}{dt} =$$

$$= \omega^2 R\left[\cos\theta + \frac{(-\sin\theta(\Lambda\sin\theta-\delta)+\cos\theta\Lambda\cos\theta)\sqrt{1-(\Lambda\sin\theta-\delta)^2}}{1-(\Lambda\sin\theta-\delta)^2} - \cos\theta(\Lambda\sin\theta-\delta)\frac{1}{2}\frac{(-2(\Lambda\sin\theta-\delta)\Lambda\cos\theta)}{\sqrt{(1-(\Lambda\sin\theta-\delta)^2)^3}}\right] =$$

$$= \omega^2 R\left[\cos\theta - \frac{\sin\theta(\Lambda\sin\theta-\delta)}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}} + \frac{\Lambda\cos^2\theta}{\sqrt{1-(\Lambda\sin\theta-\delta)^2}} + \frac{\Lambda\cos^2\theta(\Lambda\sin\theta-\delta)^2}{\sqrt{(1-(\Lambda\sin\theta-\delta)^2)^3}}\right]$$





$\sin(2\theta) = 2\sin(\theta)\cos(\theta)$

$\cos(2\theta) = \cos^2\theta - \sin^2\theta$

$$\delta = 0 : a_p = \omega^2 R\left[\cos\theta - \frac{\Lambda\sin^2\theta}{\sqrt{1-\Lambda^2\sin^2\theta}} + \frac{\Lambda\cos^2\theta}{\sqrt{1-\Lambda^2\sin^2\theta}} + \frac{\Lambda^3\cos^2\theta\sin^2\theta}{\sqrt{(1-\Lambda^2\sin^2\theta)^3}}\right] = \omega^2 R\left[\cos\theta + \frac{\Lambda\cos(2\theta)}{\sqrt{1-\Lambda^2\sin^2\theta}} + \frac{\Lambda^3\sin^2(2\theta)}{4\sqrt{(1-\Lambda^2\sin^2\theta)^3}}\right]$$

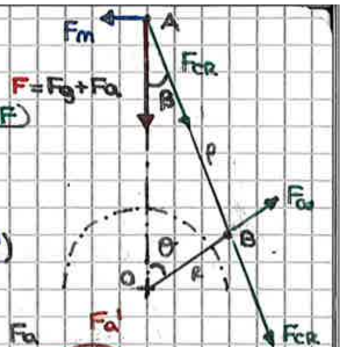
- D - FORCES

- **GAS PRESSURE FORCE (F_g)**

$F_g = [P_g(\theta) - P_0]\frac{\pi D^2}{4}$; D: BORE DIAMETER, P_0: CRANKCASE P

FORCE ON PISTON (F)

$F = F_g + F_a$



- **INERTIAL FORCE CAUSED BY MASSES WITH RECIPR. MOTION ($F_a = F_a' + F_a''$)**

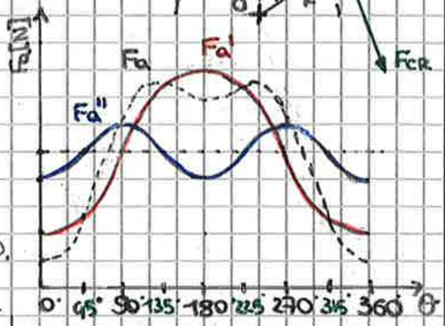
$F_a = -m_a a_p = -m_a \omega^2 R[\cos\theta + \Lambda\cos(2\theta)]$

$\begin{cases} F_a' = -m_a\omega^2 R\cos\theta : \text{FIRST-ORDER INERTIAL F.} \\ F_a'' = -m_a\omega^2 R\Lambda\cos(2\theta) : \text{SECOND-ORDER INERTIAL F.} \end{cases}$

$m_a = m_p + m_{wp} + m_{CR,a}$: PISTON GROUP MASS

F_a' IS GENERATED BY RECIPROCATING MASS ACCELERATION/D, PRODUCED BY THE CRANK ROTATING MOTION (IT IS EQUAL TO THE WHOLE INERTIA OF AN IDEAL INFINITE LENGTH C.R.)

F_a'' IS GENERATED BY AN ADDITIONAL ACCELERATION/D, PRODUCED BY THE CRANK ROTATING MOTION WHICH INCR/D. THE CONNECTING ROD OBLIQUITY ANGLE (β). ($\xi'' = 2\xi'$)



TDC (0,360°): BOTH ARE AT MAX VALUE SAME DIRECTION

BDC (180°): BOTH ARE AT MAX VALUE OPPOSITE DIRECTIONS

(90,270): $F_a' = 0$; $F_a'' = F_a''_{MAX}$

(45,135,225,315): $F_a'' = 0$

NW: BY DECREASING THE STROKE ($S = 2R\downarrow$) THE RECIPROCATING IN. F. DECREASES ($F_a\downarrow$),

- BUT, WITH THE SAME DISPLACEMENT, THE BORE DIAM. INCREASES ($D\uparrow$); THUS THE PISTON WEIGHT INCREASES ($WEIGHT_p\uparrow$). THIS SECOND EFFECT USUALLY DOES NOT AFFECT THE BENEFITS DUE TO THE STROKE DECRESE.

IF **D>S : OVER-SQUARED / SHORT STROKE ENGINES → SPORTY CARS**
- SUBJECTED TO LOWER RECIPROCATING INERTIAL FORCE ($F_a\downarrow$)
- THEY CAN ROTATE AT HIGHER SPIN SPEED
- EASIER ACCOMODATION

IF **D<S : UNDER-SQUARED / LONG STROKE ENGINES → INDUSTRIAL APPLICATIONS**
- SUBJECTED TO HIGHER RECIPROCATING INERTIAL FORCE ($F_a\uparrow$)
- PISTON TRANSLATIONAL VELOCITY REDUCES ⇒ THEY ROTATE AT LOWER SPIN SPEED ($\Lambda\downarrow V_p\downarrow \omega\downarrow$)
- PRIVILEGING HIGH TORQUE, THEREFORE HIGH POWER, AT LOW REGIME.

IF **D≈S : SQUARE ENGINE → AUTOMOTIVE APPLICATIONS**
- WE DEAL WITH THIS CONF.

- **INERTIAL FORCE CAUSED BY MASSES WITH ROTATIONAL MOTION (F_ω) (CENTRIFUGAL F.)**

$F_\omega = m_R\omega^2 R$

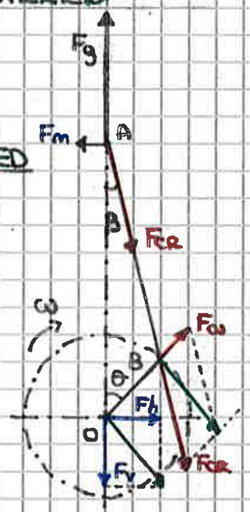
- $m_R = 2m_{cw,RED} + m_{cp} + m_{CR,R}$: CRANK GROUP MASS ($m_{cw,RED}\cdot R = m_{cw}\cdot R_{cw}$)

WHERE $m_{cw,RED} = m_{cw}\frac{R_{cw}}{R}$: CRANK WEB MASS (OR CHEEK MASS) REDUCED TO R

NW: THE CENTRIFUGAL INERTIAL FORCE CONSTANTLY PASSES THROUGH THE CRANKSHAFT ROTATION AXIS; THEREFORE IT DOES NOT INFLUENCE THE TORQUE (M).

- D - FORCES (F_h F_v) ACTING ON THE MAIN BEARING, CENTERED

$$\begin{cases} F_h = F_\omega \sin\theta + F_{CR} \sin\beta = F_\omega \sin\theta + F_m \\ F_v = F_{CR}\cos\beta - F_\omega \cos\theta = F - F_\omega\cos\theta = F_g + F_a - F_\omega\cos\theta \end{cases}$$



- D - FORCES (F_H, F_V) AND MOMENT (M_e) ON CYL. BLOCK, CENTERED

$$\begin{cases} F_H = F_h - F_m = F_\omega \sin\theta \\ F_V = F_v - F_g = F_a - F_\omega\cos\theta \end{cases}$$

- F_H DEPENDS ONLY ON CENTRIFUGAL FORCE (F_ω)
- F_V DEPENDS ON BOTH CENTR. (F_ω) AND RECIPR. (F_{CR}) FORCES
- F_V DOES NOT DEPEND ON GAS FORCE (F_g) (BECAUSE F_g ACTS IN EQUAL MANNER BOTH ON CYL. HEAD AND ON PISTON)

$M_e = F_m \cdot x(\theta) = F_m(\ell\cos\beta + R\cos\theta)$: COUNTER CLOCKWISE EQ. MOMENT ON CYL. BLOCK ($\circlearrowleft$)

DEVELOPING ITS EXPRESSION:

$$M_e = F_m \cdot x(\theta) = F_m(\ell\cos\beta + R\cos\theta) = F\tan\beta(\ell\cos\beta + R\cos\theta) = F(\ell\cdot\sin\beta + R\cos\theta\tan\beta)$$

$$= F\left(\ell\Lambda\sin\theta + \frac{R\cos\theta\,\Lambda\sin\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}\right) = F\cdot R\left(\sin\theta + \frac{\Lambda\sin\theta\cos\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}\right)$$

NEGLECTING $\Lambda^2\sin^2\theta$:

$M_e = F\cdot R(\sin\theta + \Lambda\sin\theta\cos\theta) = F\cdot R\left(\sin\theta + \frac{\Lambda\sin(2\theta)}{2}\right)$: APPROX C.C. EQ. MOMENT ($\circlearrowleft$)

$M_e = M$: IT IS **EQUAL** AND **OPPOSIT** TO THE INSTANTANEOUS SINGLE CRANK TORQUE (M^*)

IT DEPENDS BOTH ON F_g AND F_{CR}

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- FOR THE FLYWHEEL DIMENSIONING:

$$J_{FL} = \frac{\Delta E}{\Delta \cdot \omega^2} \rightarrow J_{FLV} = \frac{J_{FL}}{\rho} = s \cdot I_{FL} = s \cdot \frac{\pi}{32}(D^4 - d^4) \Rightarrow s = \frac{J_{FLV}}{I_{FL}}$$

- THE FLYWHEEL DIMENSIONING IS A ITERATIVE PROCEDURE. WE START WITH A FIRST ATTEMPT FLYWHEEL DIMENSIONING. WE PROCEED WITH THE ANALYSIS OF THE OTHER COMPONENTS, AND, IN PARTICULAR, OF THE CRANKSHAFT. WE COMPUTE THE LEVEL OF TORSIONAL VIBRATIONS; IF IT DOES NOT CORRESPOND TO THE SPIN SPEED (ω) THAT WE INITIALLY ASSUMED, WE HAVE TO COME BACK TO RE-SIZE THE FLYWHEEL ACCORDING TO THIS NEW COMPUTED SPIN SPEED (ω').

$\left(I_{FL} = \frac{\pi}{32}(D^4 - d^4)\right.$: MOMENT OF INERTIA OF AREA$\left.\right)$

E3 CONNECTING ROD

- DG - STEM

1st STEP [COMPRESSIVE STATE]

$$A_{CR} = \frac{F \cdot SF}{R_{p0.2}} \quad \left(\Rightarrow F = \frac{A_{CR} R_{p0.2}}{SF} \right)$$

$$F = MAX \left(Fg_{TDC} ; Fa_{BDC} \right)$$

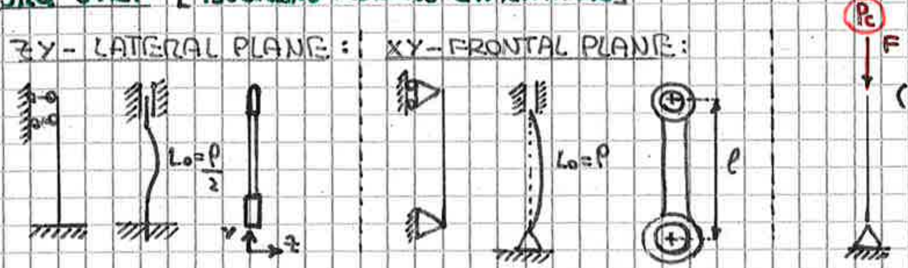
2nd STEP [TENSILE STATE]

$$\sigma_t = \frac{Fa_{MAX}}{A_{CR}} = \frac{m_{A} \omega^2 R (1+\Lambda)}{A_{CR}} \leq \sigma_{ADM} \quad (\sigma_t^{STEM})$$

$$Fa_{MAX} = Fa_{TDC}$$

3rd STEP [BUCKLING MODELS EVALUATION]

ZY - LATERAL PLANE : | XY - FRONTAL PLANE :



$F = MAX (Fg_{TDC}, Fa_{BDC}) < P_c$

(COMPRESSIVE ACTION)

$$P_c = \frac{\pi^2 E I_{CR}}{L_0^2}$$: CRITICAL BUCKLING LOAD (EULER EQ.) ; SINCE $L_{0xy} > L_0$ → XY-F. PLANE EV.

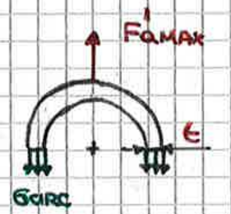
$$F < P_c \iff \frac{A_{CR} R_{p02}}{SF} \leq \frac{\pi^2 E I_{CR}}{\ell^2} \Rightarrow \frac{A_{CR}}{I_{CR}} \leq \frac{\pi^2 E\, SF}{R_{p02}\, \ell^2}$$

- DG - SMALL END

$$\sigma_{CIRC} = \frac{Fa'_{MAX}}{2bt} = \frac{(m_{p} + m_{WP}) \omega^2 R (1+\Lambda)}{2bt}$$

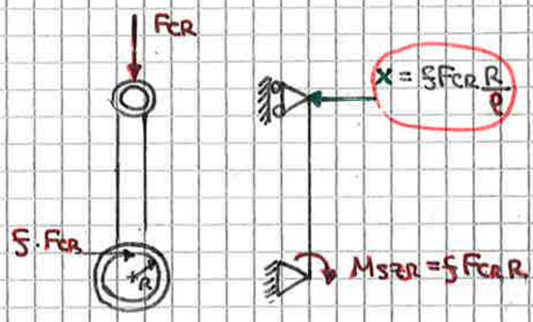
$Fa'_{MAX} = Fa'_{TDC}$ [TENSILE EFFECT]

BEING A THIN TUBE ⇒ $\sigma_{CIRC} \sim CONST$



PREVIEW: (VG) EQUIVALENT STRESS ACTING ON THE SMALL EYE (σ_{EQ}^{SE})

$$\sigma_{EQ}^{SE} = \sigma_{CIRC} + \sigma_{FIT} + \sigma_{CURV} \leq \sigma_{ADM}$$ (AT A DISTANCE $\propto r_{CURV} \propto$ SE GEOMETRY)



BENDING MOMENT DUE TO THE SEIZURE

$$M_{SZR} = \xi F_{CR} \cdot R = \xi \frac{F}{\cos\beta} \cdot R = M_{SZR\,MAX}$$

EVALUATED AT $y_{MWPL\,MAX} = \ell - \ell/\sqrt{3}$

$$M_{SZR}(y_{MWPL\,MAX}) = M_{SZR}\left(\frac{\ell - y_{MWPL\,MAX}}{\ell}\right)$$

SEIZURE BENDING STRESS (σ_{SZR}):

$$\sigma_{SZR} = \frac{M_{SZR}}{W_{CR}}\frac{(\ell - y_{MWPL\,MAX})}{\ell}$$

(RECALLING THE TENSILE STRESS (σ_t)):

$$\sigma_t = \frac{F_{a\,MAX}}{A_{CR}} = \frac{m_a \omega^2 R(1+\lambda)}{A_{CR}}$$

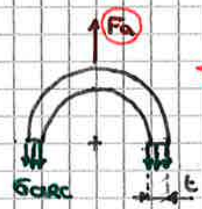
EQUIVALENT STRESS FOR THE STEM (σ_{EQ}^{STEM}):

$$\sigma_{EQ}^{STEM} = \sigma_t + \sigma_{WPL} + \sigma_{SZR} \leq \sigma_{ADM} \qquad (\text{AT } y = \ell - \ell/\sqrt{3})$$

FOR FATIGUE VERIFICATION:

$$\sigma_m = 0 \quad \sigma_a = \sigma_{EQ}^{STEM}$$

- VG - SMALL EYE



THIN TUBE MODEL

(RECALLING THE CIRCUMF. STRESS (σ_{CIRC})):

$$\sigma_{CIRC} = \frac{F_a}{2bt} = \frac{(m_P + m_{WP})\omega^2 R(1+\lambda)}{2bt}$$

FITTING STRESS (σ_{FIT})

$$p = \frac{i_{MAX} - \Delta i^T}{D_c(\delta_{SE} + \delta_b)} = \dots$$

$\delta_{SE} = \dots$

$\delta_b = \dots$

THICK TUBE MODEL

(LAMÈ THEORY)

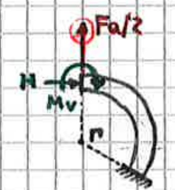
R: RADIAL COORDINATE

$$\sigma_{FIT} = p \cdot \frac{D_E^2 + D_C^2}{D_E^2 - D_C^2} \qquad \sigma_{FIT} = A + \frac{B}{R^2} = \sigma_{CIRC}\,;\quad \sigma_{rad} = A - \frac{B}{R^2}$$

CURVATURE STRESS (σ_{CURV}):

$$H \propto F_a\,\xi(\Gamma) \qquad H = \dots$$

$$M_V \propto F_a \frac{D_{SE}}{2} g(\Gamma) \qquad M_V = \dots$$



$$M_{MAX} = M_V + H\frac{D_{SE}}{2}\cos(\gamma) + \frac{F_a}{2}\frac{D_{SE}}{2}\sin(\gamma)$$

$$\sigma_{CURV} = \frac{M_{MAX}}{W_{SE}}$$

EQUIVALENT STRESS FOR THE SMALL END (σ_{EQ}^{SE})

$$\sigma_{EQ}^{SE} = \sigma_{CIRC} + \sigma_{FIT} + \sigma_{CURV} \leq \sigma_{ADM} \qquad (\text{AT A DISTANCE} \propto \Gamma_{CURV} \propto \text{SE GEOMETRY})$$

EG CRANKSHAFT

DESIGN AND VERIFICATION GUIDELINES

MAIN JOURNAL AND CRANKPIN JOURNAL DIAMETERS (d_{mj}, d_{cpj}):

BY DEFINITION m. OF CYL., D, S, $u = \frac{2Sm}{60}$ ARE DEFINED (...)

IN FIRST APPROX: $d_{mj} = d_{cpj}$

MAIN JOURNAL AND CRANKPIN JOURNAL LENGTHS (L_{mj}, L_{cpj}):

$$\begin{cases} L_{cpj} = \dfrac{F}{d_{cpj}\,p_{ADM}} \\ L_{mj} = \dfrac{F}{2 d_{cpj}\,p_{ADM}} \end{cases} \quad \text{WITH} \quad \begin{cases} F = F_g + F_a \\ p_{ADM} = 18 \div 25\,[MPa] \quad \propto f(\text{BUSHING MATERIAL}) \end{cases}$$

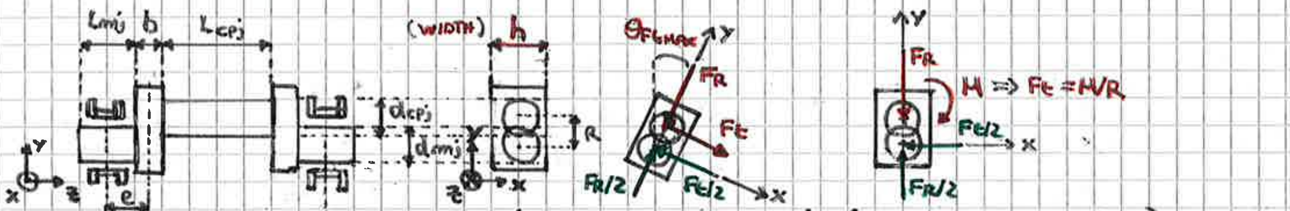
– DG – $F_{t\,MAX}$ CONFIGURATION

WE WANT TO DEFINE PARAMETERS a, e, b WE CONSIDER THE CONF. OF MAX TANG. F. ($F_{t\,MAX}$) GENERATED UNDER THE EFFECT OF MAX GAS PRESS. F. ($F_{g\,MAX}$) (MAX T REGIME)

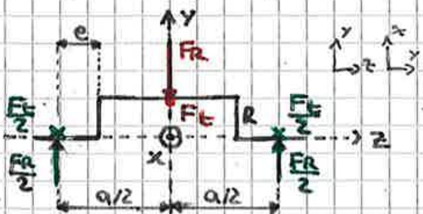
(σ_{ADM} = 60 [MPa]: CAST IRON; σ_{ADM} = 100 [MPa]: STEEL)

– VG – TDC CONFIGURATION

WHILE, FOR THE VERIFICATION WE CONSIDER THE TDC CONFIG. WHERE F_g DOES NOT GENERATE TORQUE AND F_t DERIVES FROM THE MODULATION OF M GENERATED BY THE OTHER CRANKS



($F_{t\,MAX}$ CONFIGURATION) (TDC CONFIGURATION)



$$\begin{cases} F_t = F_{CR} \sin(\theta+\beta) \\ F_R = F_{CR} \cos(\theta+\beta) \\ F_{CR} = \dfrac{F}{\cos\beta} = \dfrac{F_g + F_a}{\cos\beta} \end{cases}$$

THE CRANKPIN JOURNAL IS LOADED BY (M_x, M_y), (M_z):

$M_x = \frac{F_R}{2}\frac{a}{2}$; $M_y = \frac{F_t}{2}\frac{a}{2}$; $M_z = \frac{F_t}{2} \cdot R$

$\sigma_b = \frac{M_b}{W_b} = \frac{\sqrt{M_x^2 + M_y^2}}{W_b} = \frac{a}{4}\frac{\sqrt{F_R^2 + F_t^2}}{W_b}$; $\tau = \frac{M_z}{W_t} = \frac{F_t R}{2 W_t}$

$\sigma_{EQ} = \sqrt{\sigma_b^2 + 3\tau^2} \leq \sigma_{ADM} \rightarrow$ (a)

THE MAIN JOURNAL IS LOADED BY (M_x, M_y), (M_z):

$M_x = \frac{F_R}{2} e$; $M_y = \frac{F_t}{2} e$; $M_z = \frac{F_t}{2} R$

$\sigma_b = \frac{M_b}{W_b} = \frac{\sqrt{M_x^2 + M_y^2}}{W_b} = \frac{e}{2}\frac{\sqrt{F_R^2 + F_t^2}}{W_b}$; $\tau = \frac{M_z}{W_t} = \frac{F_t R}{2 W_t}$

$\sigma_{EQ} = \sqrt{\sigma_b^2 + 3\tau^2} \leq \sigma_{ADM} \rightarrow$ (e)

THE CHEEK IS LOADED BY (N), (M_x, M_z):

$N = \frac{F_R}{2}$; $M_x = \frac{F_R}{2} e$; $M_z = \frac{F_t}{2}$

$\sigma_{EQ} = \frac{N}{A} + \frac{M_x}{W_x} + \frac{M_z}{W_z} = \frac{N}{bh} + \frac{M_x}{hb^2} + \frac{M_z}{bh^2} \leq \sigma_{ADM} \rightarrow$ (b)

THERMAL VERIFICATION

$\left[\frac{kg\,m}{cm^2\,s}\right]$ $\frac{p_{Mi} \frac{\pi D^2}{4}}{2 d_{cpj} L_{mj}} \frac{\pi m}{30} d_{cpj} \leq 15 \div 25\,[MPa]$

STRUCTURAL VERIFICATION:

THE CRANKPIN J. IS LOADED BY (M_x), (M_z)

$M_x = \frac{F_R}{2}\frac{a}{2}$; $M_z = \frac{F_t}{2} R$ (NO MORE M_y)

$\sigma_b = \frac{M_x}{W_b} = \frac{F_R a}{4 W_b}$; $\tau = \frac{M_z}{W_t} = \frac{F_t R}{2 W_t}$

$\sigma_{EQ} = \sqrt{\sigma_b^2 + 3\tau^2} \leq \sigma_{ADM}$

THE MAIN J. IS LOADED BY (M_x), (M_z)

$M_x = \frac{F_R}{2} e$; $M_z = \frac{F_t}{2} R$ (NO MORE M_y)

$\sigma_b = \frac{M_x}{W_b} = \frac{F_R e}{2 W_b}$; $\tau = \frac{M_z}{W_z} = \frac{F_t R}{2 W_z}$

$\sigma_{EQ} = \sqrt{\sigma_b^2 + 3\tau^2} \leq \sigma_{ADM}$

THE CHEEK IS LOADED BY (N), (M_x, M_z):

$N = \frac{F_R}{2}$; $M_x = \frac{F_R}{2} e$; $M_z = \frac{F_t}{2} R$ (SAME)

$\sigma_{EQ} = \frac{N}{bh} + \frac{M_x}{hb^2} + \frac{M_z}{bh^2} \leq \sigma_{ADM}$

CONSIDERING THE MULTI-CYLINDER CONDITION:

$$M_j(\theta) = M_{gj}(\theta) + M_{aj}(\theta) = \sum_{k=1}^{20} \left(M_{gk}\, e^{i(k\omega' t + \varphi_{gk} + \delta_{kj})} + \omega^2 M_{ak}\, e^{i(k\omega' t + \varphi_{ak} + \delta_{kj})} \right)$$ ¡!

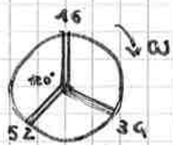
$\delta_{kj} = ORD \cdot \psi$: PHASE SHIFT (BETWEEN E.T.H.) ; ψ : ANGULAR PHASE SHIFT

PHASE DIAGRAM OF THE CRANKS:

$$\psi = \begin{cases} \frac{4\pi}{z} & : 4S\text{-}E. \\ \frac{2\pi}{z} & : 2S\text{-}E. \end{cases}$$

EXAMPLE: 4S, 6-CYL.E ; F.O.: 1,5,3,6,2,4

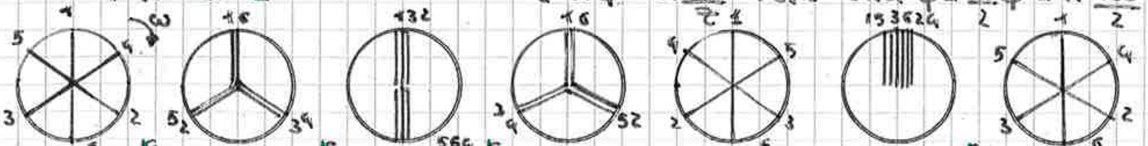
$\psi = \frac{4\pi}{6} = \frac{2}{3} \cdot 180° = 2 \cdot 60° = 120°$



PHASE DIAGRAM OF THE HARMONICS:

$$ORD = \begin{cases} \frac{k}{2} & : 4S\text{-}E \\ k & : 2S\text{-}E \end{cases} \qquad \delta_{kj} = ORD \cdot \psi = \begin{cases} \frac{k}{2}\psi = \frac{k}{2}\frac{4\pi}{z} \\ k\psi = k\frac{2\pi}{z} \end{cases}$$

IN THE EXAMPLE: $\delta_{jk} = ORD \cdot \psi = \frac{k}{2}\psi = k \cdot \frac{120°}{2} = k \cdot 60°$



$k=1$	$k=2$	$k=3$	$k=4$	$k=5$	$k=6$	$k=7$
ORD = 0.5	ORD = 1	ORD = 1.5	ORD = 2	ORD = 2.5	ORD = 3	ORD = 3.5
$\delta_{jk} = 60°$	$\delta_{jk} = 120°$	$\delta_{jk} = 180°$	$\delta_{jk} = 240°$	$\delta_{jk} = 300°$	$\delta_{jk} = 360°$	$\delta_{jk} = 420°$
GROUP 1	GROUP 2	GROUP 3			GROUP 4	

- m OF DIFFERENT GROUPS = $\begin{cases} \frac{z+2}{2} & : 4S - \text{EVEN } m. \text{ OF CYLS} - E. \\ \frac{z+1}{2} & : 4S - \text{ODD } m \text{ OF CYLS} - E. \end{cases}$ | IN THE EXAMPLE: $\frac{6+2}{2} = 4$ DIFF. GROUPS
- IF ORD = MULTIPLE OF $\frac{z}{2}$ ⇒ "MAJOR ORDERS" ($z/2 = 6/2 = 3$) ($(z+2)/2 = (6+2)/2 = 4$ DIFF GROUPS)
- DANGEROUS HARMONICS : RESONANT H., MAJOR ORDER H., $\frac{\text{MAJOR ORDER}}{2}$ H. (SOMETIMES)

FORCED RESPONSE (IN RESONANT CONDITION):

$\{\phi\}_k = \alpha_k \{\phi\}_{RES}$: FORCED RESPONSE AMPLITUDES

$$\alpha_k = \frac{\sqrt{\left(\{\phi\}^T_{RES}\{M_{kj}\cos\delta_{kj}\}\right)^2 + \left(\{\phi\}^T_{RES}\{M_{kj}\sin\delta_{kj}\}\right)^2}}{k \cdot \omega' \cdot C_{EQ} \sum_{v=1}^{z+2} \{\phi\}^T_{RES}\{\phi\}_{RES}}$$: PROPORTIONALITY FACTOR ¡!

$C_{EQ} = 2 \cdot 10^5 \cdot A R^2$: VISCOUS DAMPING $[Nms/Rad]$

$2 \cdot 10^5$: CRANK NUMERICAL COEFF. $[Ns/Rad/m^3]$

TOTAL TORSIONAL SHEAR STRESS (ACTING ON EACH TORSIONAL BAR):

$$\tau_s = \tau_{STAT,s} + \tau_{DYN,s} = \frac{s \cdot A_0}{W_{t,s}} + \frac{K_s(\phi_{s+1} - \phi_s)}{W_{t,s}}$$, S : SECTIONS OF INTEREST

$$\tau_3 = \tau_{STAT,3} + \tau_{DYN,3} = \frac{3 \cdot A_0}{W_{t,3}} + \frac{K_{EQ}(\phi_4 - \phi_3)}{W_{t,3}}$$: T.S.S. IN SECTION 3

$$\{\phi\}_k = \begin{Bmatrix} \phi_1 \\ \phi_2 \\ \phi_3 \\ \phi_4 \\ \phi_5 \\ \phi_6 \end{Bmatrix}_k$$: FORCED RESPONSE AMPLITUDES

2D REYNOLDS EQUATION

$$\frac{\partial}{R_j^2 \partial\alpha}\left(h^3 \frac{\partial P}{\partial \alpha}\right) + \frac{\partial}{\partial x}\left(h^3 \frac{\partial P}{\partial x}\right) = 6\eta u \frac{\partial h}{R_j \partial \alpha} + 12\eta \frac{\partial h}{\partial t}$$

; η: KINEMATIC VISCOSITY

$u = \frac{1}{2}\omega R_j$: SPEED ON ENTRAINING MOTION

2D REYNOLDS DIFFERENTIAL EQ. → FINITE DIFFERENCE METHOD → ALGEBRAIC EQ. + B.C.:

i : ROW
j : COLUMN

$$\frac{h^3_{i+\frac{1}{2},j}\frac{P_{i+1,j}-P_{i,j}}{\Delta x} - h^3_{i-1/2,j}\frac{P_{i,j}-P_{i-1,j}}{\Delta x}}{\Delta x} + \frac{1}{R_j^2}\left(\frac{h^3_{i,j+1/2}\frac{P_{i,j+1}-P_{i,j}}{\Delta x} - h^3_{i,j-1/2}\frac{P_{i,j}-P_{i,j-1}}{\Delta x}}{\Delta x}\right) =$$

$$6\eta u \frac{h_{i+1/2,j} - h_{i-1/2,j}}{\Delta x} + 12\eta\omega \frac{h_{i,j}^{\theta+\Delta\theta} - h_{i,j}^{\theta}}{\Delta\theta}$$

BOUNDARY CONDITIONS (BC):

$$\begin{cases} P = P_{ATM} & x = \pm L_x/2 \\ \partial P/\partial\alpha = 0 & \alpha = 0, \pi \end{cases}$$

DIMENSIONLESS FORM OF 2D REYNOLDS EQ:

...

DIMENSIONLESS PARAMETERS:

...

...

...

HYDRODYNAMIC PRESSURE:

...

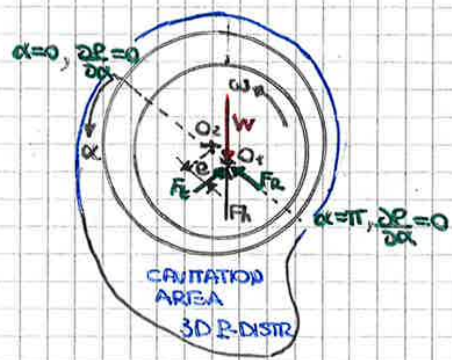
INSTANTANEOUS FORCE EQUILIBRIUM / BALANCE → ECCENTRICITY (e) → L.O. THICK. ($h(\alpha)$)
→ TIME EVOLUTION OF THE HYDRODYNAMIC PRESSURE (P) IN THE AXIAL (x) AND CIRCUM (α) NODES

$W - F_h = 0$

$F_h = \sqrt{F_R^2 + F_t^2}$

$F_R = \int_0^{\pi}\int_{-L_x/2}^{L_x/2} P(x,\alpha) R \cos\alpha \, dx\, d\alpha$

$F_t = \int_0^{\pi}\int_{-L_x/2}^{L_x/2} P(x,\alpha) R \sin\alpha \, dx\, d\alpha$



E8 CYLINDER BLOCK

- DG - WALL

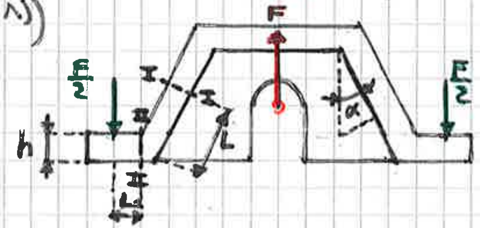
- DG - WALL - SCREW C. - SECTION I-I ($\sigma_{EQ}^{I} = \sigma_t^{I} + \sigma_b^{I} \leq \sigma_{ADM}$)

$\begin{cases} 20\,[MPa]: \text{C.I., Al-ALLOY} \\ 40\,[MPa]: \text{CAST STEEL} \\ 60\,[MPa]: \text{WELD STEEL} \end{cases}$

$$F = MAX(F_{g\,MAX}, F_{a\,MAX}) = MAX\left(\frac{P_{g\,MAX}\pi D^2}{4},\, m_a \omega^2 R(1+\lambda)\right)$$

$F_t^{I} = \frac{F}{2}\cos\alpha$ | $\sigma_t^{I} = \frac{F_t^{I}}{A^{I}}$ | $\sigma_{EQ}^{I} = \sigma_t^{I} + \sigma_b^{I} \leq \sigma_{ADM}$

$M_b^{I} = \frac{F}{2} L \sin\alpha$ | $\sigma_b^{I} = \frac{M_b^{I}}{W_b^{I}}$



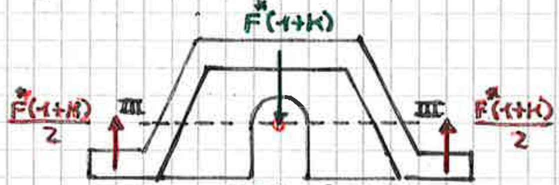
- DG - WALL - SCREW C. - SECTION II-II ($\sigma_b^{II} \leq \sigma_{ADM}$)

$M_b^{II} = \frac{F}{2} L_1$ | $\sigma_b^{II} = \frac{M_b^{II}}{W_b^{II}} \leq \sigma_{ADM}$

- DG - WALL - STUD C. - SECTION III-III ($\sigma_c^{III} \leq \sigma_{ADM}$)

$\begin{cases} 40\,[MPa]: \text{C.I. Al-ALLOY} \\ 80\,[MPa]: \text{CAST STEEL} \end{cases}$

$$\sigma_c^{III} = \frac{F^*(1+K)}{A^{III}} \leq \sigma_{ADM} \quad ; \quad F^* = P_{MAX}\frac{\pi d_g^2}{4}$$



- DG - CAP

- DG - CAP - BOTH FOR SCREW/STUD C. - SECTION I-I ($\sigma_b^{I} \leq \sigma_{ADM}$)

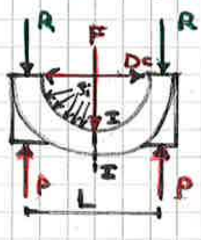
$\begin{cases} 20\,[MPa]\ \text{CAST IRON} \\ 60\,[MPa]\ \text{CAST STEEL} \\ 80\,[MPa]\ \text{FORGED STEEL} \end{cases}$

$$F = F_{g\,MAX} = P_{g\,MAX}\frac{\pi D^2}{4} \ ; \ P = \frac{F(1+K)}{2} \ ; \ R = P - \frac{F}{2} = \frac{F(1+K)}{2} - \frac{F}{2} = \frac{FK}{2}$$

$$M_b^{I} = (P-R)\frac{L}{2} - \frac{1}{2}\sum_{i=1}^{\infty} f_i x_i = (P-R)\frac{L}{2} - \frac{1}{2} F \frac{\pi D_c^2/4}{\pi D_c} =$$

$$= (P-R)\frac{L}{2} - \frac{1}{8} F D_c = \left[\frac{F(1+K)}{2} - \frac{FK}{2}\right]\frac{L}{2} - \frac{F D_c}{8} = \frac{F}{2}\frac{L}{2} - \frac{F D_c}{8} = \frac{F}{4}\left(L - \frac{D_c}{2}\right)$$

$$\sigma_b^{I} = \frac{M_b^{I}}{W_b^{I}} = \frac{F/4\,(L - D_c/2)}{W_b^{I}} \leq \sigma_{ADM}$$

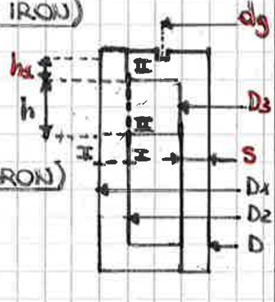


- DG - LINER (SUPPORT, WALL, FLANGE)

- DG - SUPPORT - SCREW C. - SECTION I-I ($\sigma_t^{I} \leq \sigma_{ADM}$ (20 [MPa] CAST IRON))

$$\cdot F_t^{I} = F_{g\,MAX} = P_{g\,MAX}\frac{\pi D^2}{4} \ ; \ A_I = \frac{\pi(D_1^2 - D_2^2)}{4}$$

$$\sigma_t^{I} = \frac{F_t^{I}}{A^{I}} = \frac{P_{g\,MAX} D^2}{(D_1^2 - D_2^2)} \leq \sigma_{ADM}$$



- DG - SUPPORT - STUD C. - SECTION I-I ($\sigma_t^{I} \leq \sigma_{ADM}$ (20 [MPa] CAST IRON))

$$\cdot F_t^{I} = F^*(1+K) \ ; \ F^* = P_{MAX}\frac{\pi d_g^2}{4} \ ; \ A_I = \frac{\pi(D_1^2 - D_2^2)}{4} \ ; \ (A_{II} = h \cdot \pi D_2)$$

$$\sigma_t^{I} = \frac{F_t^{I}}{A^{I}} = \frac{F^*(1+K)}{A^{I}} = \frac{P_{MAX}(1+K) d_g^2}{(D_1^2 - D_2^2)} \leq \sigma_{ADM}$$

- DG - SUPPORT - STUD C. - SECTION II-II ($\sigma_{EQ}^{II} = \sqrt{3}\,\tau^{II} \leq \sigma_{ADM}$ (20 [MPa] CAST IRON))

$$\tau^{II} = \left(\frac{F_t^{I}}{A^{II}}\right) = \frac{F^*(1+K)}{A^{II}} = \frac{F^*(1+K)}{h \cdot \pi D_2} = \frac{P_{MAX}(1+K) d_g^2}{4 h \cdot D_2}$$

$$\sigma_{EQ}^{II} = \sqrt{3}\,\tau^{II} \leq \sigma_{ADM}$$

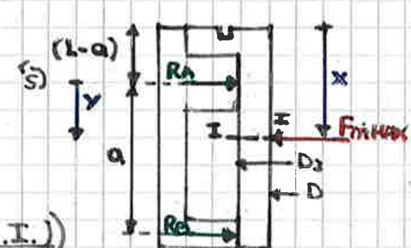
- DG - WALL - //(F_m)// - SECTION I-I ($\sigma_b^{I} \leq \sigma_{ADM}$ (20 [MPa] CAST IRON))

$$\circlearrowleft S) \ + R_B a - F_{m\,MAX} Y = 0 \qquad R_B = F_{m\,MAX}\frac{Y}{a}$$

$$R_B = F_{m\,MAX}\frac{Y}{a} = F_{m\,MAX}\frac{[x - (L-a)]}{a} = F_{m\,MAX}\frac{(x - L + a)}{a}$$

$$M_b^{I} = R_B(L-x) = F_{m\,MAX}\frac{(x - L + a)}{a}(L - x) \ ; \ W_b^{I} = \frac{\pi(D_3^4 - D^4)}{64\, D_3/2}$$

$$\sigma_b^{I} = \frac{M_b^{I}}{W_b^{I}} = \frac{32 D_3 F_{m\,MAX}(x - L + a)(L - x)}{a\pi(D_3^4 - D^4)} \leq \sigma_{ADM}$$



- DG - WALL - BOTH SCREW/STUD C. ($\sigma_{EQ}^{VM} \leq \sigma_{ADM}$ (20 [MPa] C.I.))

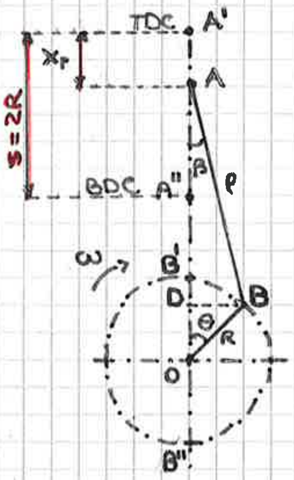
$$\begin{cases} \sigma_{CIRC} = \frac{P_i R_i}{S_i} = \frac{P_{MAX} D}{2S} \\ \sigma_{RAD} = -P_i = -P_{MAX} \\ \sigma_{AX} = \frac{E\alpha\Delta T}{2(1-\nu)} \end{cases}$$

(THIN TUBE MODEL)

$$\sigma_{EQ}^{VM} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_C - \sigma_R)^2 + (\sigma_C - \sigma_A)^2 + (\sigma_R - \sigma_A)^2} \leq \sigma_{ADM}$$

KINEMATICS

- K - PISTON DISPLACEMENT, CENTERED

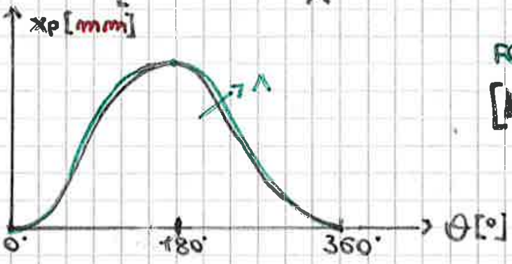


$\overline{DB} = \rho \sin\beta = R\sin\theta$ | $\Lambda = R/\rho = 0.20 \div 0.35$: ELONG. RATIO

$\sin\beta = \frac{R}{\rho}\sin\theta = \Lambda\sin\theta$; $\cos\beta = \sqrt{1-\sin^2\beta} = \sqrt{1-\Lambda^2\sin^2\theta}$

$x_p = A'O - AO = (R+\rho) - [\rho\cos\beta + R\cos\theta] = R(1-\cos\theta) + \rho(1-\cos\beta) =$

$= R\left[(1-\cos\theta) + \frac{1}{\Lambda}\left(1-\sqrt{1-\Lambda^2\sin^2\theta}\right)\right]$: PISTON DISPLACEMENT

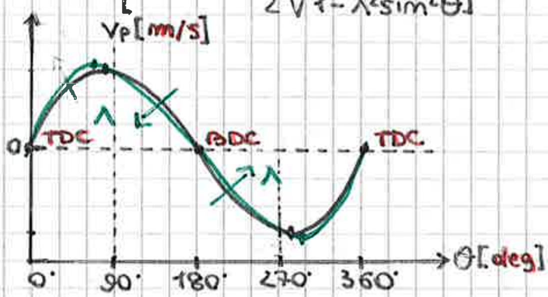


FOR COPISTERY:
[NOT TO SCAN]

- K - PISTON VELOCITY, CENTERED

$V_P = \frac{dx_P}{dt} = \frac{dx_P}{d\theta}\frac{d\theta}{dt} = \omega\frac{dx_P}{d\theta} = R\omega\left[\sin\theta - \frac{1}{\Lambda}\frac{1}{2}\frac{(-\Lambda^2)\,2\sin\theta\cdot\cos\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}\right] = R\omega\left[\sin\theta + \frac{\Lambda\sin\theta\cos\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}\right]$

$= \omega R\left[\sin\theta + \frac{\Lambda\sin(2\theta)}{2\sqrt{1-\Lambda^2\sin^2\theta}}\right]$; PISTON VELOCITY ; $V_P = \omega R\left[\sin\theta + \frac{\Lambda\sin(2\theta)}{2}\right]$: APPROXIM. PIST. VEL.



$V = V_{MAX}$ WHEN $\frac{dV_P}{dt} = 0 \iff \frac{dV_P}{d\theta}\omega = 0 \iff$

$\omega^2 R\left[\cos(\theta) + \frac{\Lambda\cos(2\theta)\,\cancel{2}}{\cancel{2}}\right] = 0$ NOT AT HALF S.

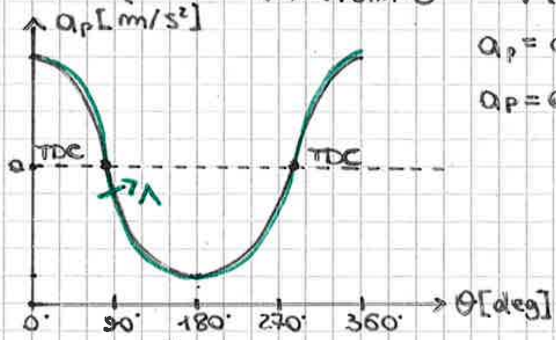
$u = \frac{2Sn}{60} = 10 \div 16$ [m/s] (22 [m/s] COMP. E)

$\frac{V_P}{u} \approx \frac{\pi}{S}R\left[\sin\theta + \frac{\Lambda\sin(2\theta)}{2}\right] = \frac{\pi}{2}\left[\sin\theta + \frac{\Lambda\sin(2\theta)}{2}\right]$

- K - PISTON ACCELERATION, CENTERED

$a_P = \frac{dV_P}{dt} = \frac{dV_P}{d\theta}\frac{d\theta}{dt} = \omega\frac{dV_P}{d\theta} = \omega^2 R\left[\cos\theta + \frac{\Lambda}{2}\left(\frac{2\cos(2\theta)\sqrt{1-\Lambda^2\sin^2\theta}}{1-\Lambda^2\sin^2\theta} - \frac{\sin(2\theta)(-\Lambda^2)\,2\sin\theta\cos\theta}{2\sqrt{(1-\Lambda^2\sin^2\theta)^3}}\right)\right]$

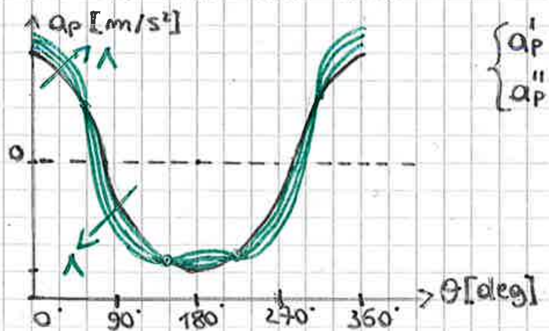
$= \omega^2 R\left[\cos\theta + \frac{\Lambda\cos(2\theta)}{\sqrt{1-\Lambda^2\sin^2\theta}} + \frac{\Lambda^3\sin(2\theta)\sin\theta\cos\theta}{\sqrt{(1-\Lambda^2\sin^2\theta)^3}}\right]$: P. ACC



$a_p = \omega^2 R\left[\cos\theta + \Lambda\cos(2\theta)\right]$: APPROX. P. ACC.

$a_P = a_{P\,MAX}$ $\frac{da_P}{dt} = 0 \iff \frac{da_P}{d\theta}\omega \iff [\sin\theta + 2\Lambda\sin(2\theta)] = 0$

$$\begin{cases} TDC\,(0°) = a_P = \omega^2 R(1+\Lambda) \\ BDC\,(180°)\; a_P = \omega^2 R(-1+\Lambda) = -\omega^2 R(1-\Lambda) \end{cases}$$



$$\begin{cases} a_P' = \omega^2 R\cos\theta & : \text{1st ORDER} \\ a_P'' = \omega^2 R\Lambda\cos(2\theta) & : \text{2nd ORDER} \quad (f'' = 2f') \end{cases}$$

– K – CONNECTING ROD, CENTERED

$\sin\beta = \Lambda \sin\theta \quad ; \quad \cos\beta = \sqrt{1-\Lambda^2 \sin^2\theta}$

$\frac{d(\sin\beta)}{dt} = \frac{d(\Lambda \sin\theta)}{dt}$

$\frac{d(\sin\beta)}{d\beta}\frac{d\beta}{dt} = \frac{d(\Lambda\sin\theta)}{d\theta}\frac{d\theta}{dt} \iff \cos\beta \frac{d\beta}{dt} = \Lambda \cos\theta\, \omega = \Lambda\omega\cos\theta$

$\omega_{CR} = \dot\beta = \frac{d\beta}{dt} = \Lambda\omega \frac{\cos\theta}{\cos\beta} = \Lambda\omega\frac{\cos\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}$: ANGULAR VELOCITY $\qquad D[\sqrt{1-\Lambda^2\sin^2\theta}] = \frac{1}{2}\frac{(-\Lambda^2)2\sin\theta\cos\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}$

$\dot\omega_{CR} = \ddot\beta = \frac{d\dot\beta}{dt} = \frac{d\dot\beta}{d\theta}\frac{d\theta}{dt} = \omega\frac{d\dot\beta}{d\theta} = \Lambda\omega^2\left[\frac{(-\sin\theta)\sqrt{1-\Lambda^2\sin^2\theta}}{1-\Lambda^2\sin^2\theta} + \frac{\cos^2\theta\,\Lambda^2\sin\theta}{\sqrt{[1-\Lambda^2\sin^2\theta]^3}}\right] =$

$= \Lambda\omega^2\left[\frac{-\sin\theta}{\sqrt{1-\Lambda^2\sin^2\theta}} + \frac{\Lambda^2\cos^2\theta\sin\theta}{\sqrt{[1-\Lambda^2\sin^2\theta]^3}}\right] = -\Lambda\omega^2\left[\frac{\sin\theta}{\sqrt{1-\Lambda^2\sin^2\theta}} - \frac{\Lambda^2\cos^2\theta\sin\theta}{\sqrt{[1-\Lambda^2\sin^2\theta]^3}}\right]$

DYNAMICS

– D – CONN. ROD – LUMPED PARAMETER SYSTEM

$$\begin{cases} m_{CR,a} + m_{CR,R} = m_{CR} \\ m_{CR,a}\, x_1 = m_{CR,R}\, x_2 \\ m_{CR,a}\, x_1^2 + m_{CR,R}\, x_2^2 + J_0 = J_{CR} \end{cases} \qquad \begin{cases} m_{CR,a} = \frac{x_2}{\ell} m_{CR} \cong (0.20 \div 0.35)\, m_{CR} \cong 0.3\, m_{CR} \\ m_{CR,R} = \frac{x_1}{\ell} m_{CR} \cong (0.80 \div 0.65)\, m_{CR} \cong 0.7\, m_{CR} \\ J_0 = J_{CR} - m_{CR}\, x_1 x_2 \cong -(0.01 \div 0.03)\, m_{CR}\frac{R^2}{\Lambda^2} \end{cases}$$

$\Delta M_i = -\dot\omega_{CR} J_0$: CORRECTIVE MOMENT OF INERTIA

– D – FORCES

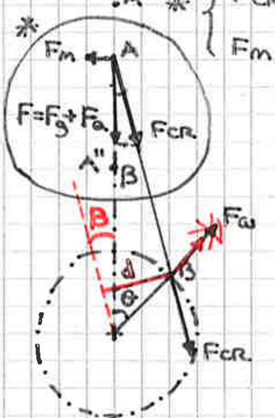
NW!

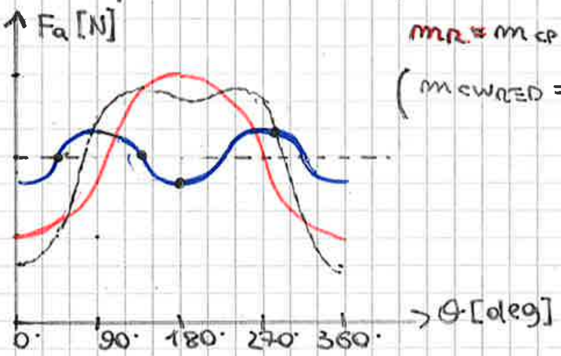
$$* \begin{cases} F_{CR} = \frac{F}{\cos\beta} \\ F_n = F\tan\beta \end{cases} \quad , F = F_g + F_a \ ; \ F_g = [P_g(\theta) - P_0]\frac{\pi D^2}{4} \ ; F_a = -m_a\omega^2 R[\cos(\theta) + \cos$$

$F_\omega = m_R\,\omega^2 R \qquad (m_a = m_p + m_{wp} + m_{CR,a}$

$m_R = m_{cp} + m_{CR,R} + 2\, m_{cw,RED}$

$\left(m_{cwRED} = m_{cw}\cdot\frac{R_{cw}}{R}\right)$





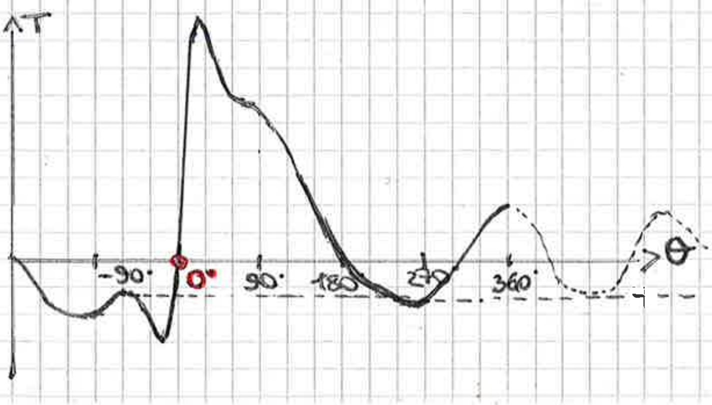
$$* \begin{cases} F_n^{OFF} < F_n^{C} \\ F_{CR}^{OFF} < F_{CR}^{C} \end{cases}$$

$M = F_{CR}\cdot d = \frac{F}{\cos\beta} R\sin(\theta+\beta) = \cancel{\frac{F}{\sqrt{1-\Lambda^2\sin^2\theta}}}\, R(\sin\theta\cos\beta + \cos\theta\sin\beta)\frac{F}{\cos\beta} =$

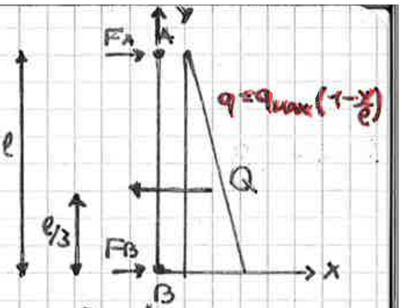
$= FR\left(\sin\theta + \cos\theta\frac{\sin\beta}{\cos\beta}\right) = FR\left(\sin\theta + \frac{\cos\theta\,\Lambda\sin\theta}{\sqrt{1-\Lambda^2\sin^2\theta}}\right) = FR\left(\sin\theta + \frac{\sin(2\theta)}{2\sqrt{1-\Lambda^2\sin^2\theta}}\right)$

$\left(M = \frac{F V_P}{\omega}\right)$

MAX VALUE STRONGLY HIGHER THAN MEAN VALUE



DEM. $M_{WPL\,MAX} = \dfrac{q_{MAX}\,\ell^2}{9\sqrt{3}}$



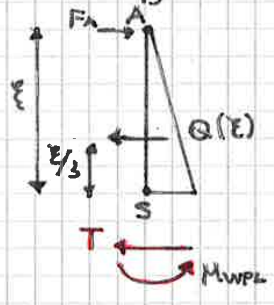
$$Q = \int_0^\ell q\,dy = \int_0^\ell q_{MAX}\left(1-\frac{y}{\ell}\right)dy = q_{MAX}\left(y-\frac{y^2}{2\ell}\right)_0^\ell = q_{MAX}\left(\ell-\frac{\ell}{2}\right) = \frac{q_M\,\ell}{2}$$

$\overset{\frown}{A})\ F_B\cdot\ell - Q(\ell-\ell/3) = 0$

$\overset{\frown}{B})\ -F_A\ell + Q\dfrac{\ell}{3} = 0$

$$\Rightarrow \begin{cases} F_B = Q\left(1-\frac{1}{3}\right) = \frac{2}{3}Q = q_{MAX}\frac{\ell}{3} \\ F_A = \frac{Q}{3} = q_{MAX}\frac{\ell}{6} \end{cases}$$

CONSIDERING:



$$Q(\xi) = \int_0^\xi q_{MAX}\frac{\xi}{\ell}\,d\xi = q_{MAX}\frac{\xi^2}{2\ell}$$

CONSIDERING: $M_{WPL} = MAX$ $\quad T = 0$

$\rightarrow)\ F_A - Q(\xi) - T = 0$

$$T = F_A - Q(\xi) = q_{MAX}\frac{\ell}{6} - q_{MAX}\frac{\xi^2}{2\ell} = q_{MAX}\left(\frac{\ell}{6}-\frac{\xi^2}{2\ell}\right) = 0$$

$$\Rightarrow \frac{\xi^2}{2\ell} = \frac{\ell}{6} \Rightarrow \xi = \sqrt{\frac{2\ell^2}{6}} = \frac{\ell}{\sqrt{3}}$$

$$\overset{\frown}{S})\ M_{WPL} + Q(\xi)\frac{\xi}{3} - F_A\xi = 0 \Rightarrow M_{WPL} = F_A\xi - Q(\xi)\frac{\xi}{3} = q_{MAX}\frac{\ell^2}{9\sqrt{3}}$$

E3

②

- CR WEIGHT **LIMIT** THE MAX SPIN SPEED AND EXHERTS IMP. TRANSV. LOAD ON.
- SHAPE & WEIGHT OPTIMIZ. (FEA)
- **FILLETS** TO MINIMIZE STR. CONC.
- I-ROD, H-ROD : DIFF. STR. DISTR., DIFF. CONSTR. PROBLEMS.
- I-ROD : CASTING / MOLDING : GOOD RESP. TO WHIPLASH AND ELAST. B. INST.
- (HIGH MOMENT OF INERTIA IN DIR. ⊥ TO REC. MOTIONS)
- H-ROD : CASTING / MELTING
- **CROSS**-MOUNTING
- SERRATION / TOOTHING
- SMALL EYE : ... THIN PIPE, **DRILLED** ⊥ TO AXIS (TO ALLOW L. ADD) ; WEDGE SHAPED
- <u>MATERIALS</u> : THE CHOICE : MASS, **STRENGTH**, ... → AS Ti-ALLOY
  - HARDENED ALLOY STEELS (34NiCrMo12), HOT FORGING (EXPENSIVE)
  - SINTERED STEEL IN METAL MATRIX COMPOSITE, Al-ALLOY WITH C-FIBERS, H. PERF. TH. PL. M. (PEEK)
  - LOST WAX ; S. FIN., PRECISE TOLER, DRAFT ANGLES, UNDERCUTS.
  - Al-ALLOY : Si-Mg MELTING ; Si-Cu MOLDING
  - SHOT-PEENT. F.R. (UP TO 25%)
  - <u>DG</u> $A_{cr} = \frac{F \cdot SF}{R_{p0.2}}$ $F = MAX(F_{g\,MAX}, \cancel{F_{a\,MAX}})$ $\sigma_t = \frac{F_{a\,MAX}}{A_{cr}} \le R_{p0.2}$
  - CROSS SECTION **SHAPE** : (LATERAL) (FRONT) ROLLER SUPPORT

  - EULER EQ. → BUCK. CR. LOAD (P_c)

$$P_c = \frac{\pi^2 E I_{cr}}{L_0^2} \qquad F = \frac{A_{cr} R_{p0.2}}{SF} \le \frac{\pi^2 E I_{cr}}{L_0^2} \Rightarrow \frac{A_{cr}}{I_{cr}} < \frac{\pi^2 E\, SF}{\rho^2 R_{p0.2}} \qquad L_0^2 = \rho^2$$

- <u>DG</u> - SMALL EYE ; (THIN TUBE)

$$\sigma_{circ} = \frac{F_{a\,MAX}}{2bt} = \frac{(m_p + m_{wp})\omega^2 R(1+\Lambda)}{2bt}$$ "BEING THIN" → $\sigma_{circ} \sim$ CONST

- <u>VERIFICATION</u>
- <u>VG - STEM</u>

$$\omega_{cr,MAX}(\theta = 90°) = \ddot{\beta}_{MAX} = \frac{\omega^2 \Lambda}{\sqrt{1-\Lambda^2}}$$

$$\int_V \ddot{\beta}_{MAX}\, dm_{cr} = \int_0^l \frac{\omega^2\Lambda}{\sqrt{1-\Lambda^2}} \rho A_{cr}(l-y)\, dy = \int_0^l q\, dy \; ; \; q_{MAX}(y=0) = \frac{\omega^2 \Lambda}{\sqrt{1-\Lambda^2}} \rho A_{cr} l$$ — F. ON THE STEM

$$M_{wpl} = \frac{q_{MAX} l^2}{9\sqrt{3}} \qquad y_{M_{WPL,MAX}} = l - \frac{l}{\sqrt{3}}$$

$$q = q_{MAX}(1 - y/l) \; ; \; Q = \int_0^l q\, dy = \int_0^l q_{MAX}\left(1 - \frac{y}{l}\right) dy \; ; \; Q(\xi) = \int_0^{\xi} q_{MAX} \frac{\xi}{l}\, dy$$

- <u>SEIZURE</u> :

$$\sigma_{szr} = \frac{M_{szr}}{W_{cr}} \frac{l - y_{MAX}}{l} \; , \; M_{szr} = R f F_{cr}$$

$$\sigma_{eq,STEM} = \sigma_t + \sigma_{wpl} + \sigma_{szr}$$

- **FATIGUE** : $\sigma_m = 0$ $\sigma_a = \sigma_{eq,STEM}$

(4)

DYNAMIC ANALYSIS

$[J]\{\ddot{\Phi}\} + [K]\{\Phi\} = \{M(\theta)\}$

$\bar{J}_{EQ} = J_C + m_{CR,R} R^2 + (m_p + m_{wp} + m_{CR,A}) \frac{8+2\Lambda^2+\Lambda^4}{16} R^2 + J_o \frac{4\Lambda^2+\Lambda^4}{8}$

$K_{EQ} = \frac{G I_P}{L_{EQ}}$ | $\{\Phi\} = \{\Phi_0\} e^{i\lambda m t}$

$M(\theta) = M_g(\theta) + M_a(\theta)$ (ACTUAL M_g VS F. SERIES)

$M_g(\theta) = P_g(\theta) \frac{\pi D^2}{4} \cdot f_1(\theta) \cdot R \cong A_0 + \sum_{k=1}^{20} A_k \cos(k\omega' t) + \sum_{k=1}^{20} B_k \sin(k\omega' t)$ $\begin{cases} \omega' = \frac{\omega}{2} \\ \omega' = \omega \end{cases}$

$M_a(\theta) = m_a \omega^2 R \, \cancel{(\ldots)} f_2(\theta) \cdot R \cong \omega^2 \sum_{k=1}^{8} A_k^* \cos(k\omega t) + \omega^2 \sum_{k=1} B_k^* \sin(k\omega t)$

$A_0 = \frac{1}{2\pi}\int_0^{2\pi} M_g \, d(\omega' t)$ $A_k = \frac{1}{\pi}\int_0^{2\pi} M_g \cos(k\omega' t) \, d(\omega' t)$; $B_k = \frac{1}{\pi}\int_0^{2\pi} M_g \sin(k\omega' t) \, d(\omega' t)$

$M_{gk} = \sqrt{A_k^2 + B_k^2}$ $\varphi_{gk} = \tan^{-1}\left(\frac{B_k}{A_k}\right)$ | $A_k^* = \frac{1}{\pi}\int_0^{2\pi} M_a \cos(k\omega t) \, d(\omega t)$

$M_g(\theta) = \sum_{k=1}^{20} M_{gk} e^{i(k\omega' t + \varphi_{gk})}$ | $\varphi_{ak} = \pi/2$ IN CASE OF CENT. LAYOUT

$M_a(\theta) = \omega^2 \sum_{k=1}^{8} M_{ak} e^{i(k\omega t + \varphi_{ak})}$

$M_j(\theta) = M_{gj}(\theta) + M_{aj}(\theta) = \sum_{k=1}^{20} \left(M_{gk} e^{i(k\omega' t + \varphi_{gk} + \delta_{kj})} + \omega^2 M_{ak} e^{i(k\omega' + \varphi_{ak} + \delta_{kj})} \right)$

$\delta_{kj} = ORD \cdot \psi$: PHASE SHIFT ; ψ = ANGULAR PHASE SHIFT

PHASE DIAGRAM OF THE CRANKS

$\begin{cases} \psi = \frac{4\pi}{z} = \frac{4 \cdot 180}{6} = 120° \\ \psi = \frac{2\pi}{z} \end{cases}$  FO: 1, 5, 3, 6, 2, 4 ORD $\begin{cases} \frac{k}{2} & 4S \\ k & 2S \end{cases}$

PHASE DIAGRAM OF THE HARMONICS

$\delta_{kj} = ord \cdot \psi = \frac{k}{2} 120 = k \cdot 60°$

n OF DIFF. GROUPS : $\begin{cases} \frac{z+2}{2} & 4S \text{ EVEN} \\ \frac{z+1}{2} & 4S \text{ ODD} \end{cases}$ $\frac{6+2}{2} = 4$

IF ord = MULTIPLE $\frac{z}{2}$ $\left(\frac{6}{2} = 3\right) \Rightarrow$ MAJOR ORDER (III)

DANG: H: RES., MAJ, MAJ/2

FORCED RESPONSE (RES. COND)

$\{\Phi\}_k = \alpha_k \{\Phi\}_{RES}$

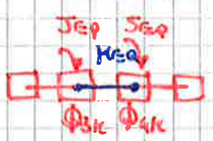
$$\alpha_k = \frac{\sqrt{\left(\{\Phi\}^T_{RES}\{M_{kj}\cos(\delta_{kj})\}\right)^2 + \left(\{\Phi\}^T_{RES}\{M_{kj}\sin(\delta_{kj})\}\right)^2}}{k\omega' C_{EQ} \cdot \sum_{\gamma=1}^{z+2} \{\Phi\}^T_{RES}\{\Phi\}_{RES}}$$

$C_{EQ} = 2 \cdot 10^5 \cdot A \cdot R^2$: DAMP. COEFF. [Nms/Rad]

$2 \cdot 10^5$: CRANK. NUM. COEFF [Ns/Rad/m³]

TOTAL TORS. SHEAR STRESS (ACTING ON EACH TORSIONAL BAR)

$\tau_s = \tau_{STAT,s} + \tau_{DYN,s} = s \frac{A_0}{W_{t,s}} + K_s \frac{(\Phi_{s+1} - \Phi_s)}{W_{t,s}}$



FEA-MBA N. SIM.

→ FORCED RESP. CORR TO MAX ENGT IS INVEST

→ THROUGH EHD SIMULATION → 3D DISTRIB. OF OIL FILM P (& OIL FILM MIN THK)

→ CAVITATION AREA BR PARABOLIC CIRCUMFERENTIAL - ASIMMETRIC DISTRIB.

CR 10 min
→ CB 9 h
BR 6 h ← OVER EST. P. DISTR., MISALIGN
BE 60 h UNDER EST. OIL FILM THICKNESS

- CLOSED DESIGN: MODIFIED DIE CAST PROCESS
- INTERNAL SURFACE FINISHING → GOOD TRIB. COUPLING (WEAR)
  → GOOD SEALING EFFECT (↓ LUB. CONS; BLOW-BY)
- DEFORM. OF LONGITUDINAL CYL. SURF. – MECH STRESS: TIGHT, Pa
  – THERM STRESS: HEAT, DUR. COMB.
- 8-9 [μm] (BASIC) 2-3 [μm] (FINE)
- CROSS RIFLING / DASHED HELICAL RIFLING
- NUMERICAL CFD → WATER JACKET

E9 *

- TO INCR. B. N. FREQ. OF POWERTRAIN; TO STIFFEN THE LINK
- IN CASE OF STR. F. → STIFFENING RIBS
- SANDWICH SOL. → TO MINIMIZE ACOUSTIC EM.
- DIE CAST / SHELL CAST Al-ALLOY
- GOALS NVH PERFORM.; ENGINE WEIGHT SAVING
- NUMERICAL OPT. METHODOLOGY
- ± 20% → PASS

E11

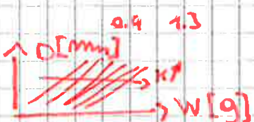
- IN MODERN DIESEL ENGINE ...
- MATERIALS AISI 400
- G. OPT TO REDUCE THERMAL STRAIN
- BEAMS, ..., AND ANALYZING THE DEF. CONF
- W.C. ON TEST BENCH: CYCLIC NON-ISO-T LOADS (800, 1000 [°C])
- DIRECT GAS LEAKAGES TEST
- MAT. BEHAV: ISO-T (HIGH T) LCF ... INTER/TRANS-GRAN. CRACK PROP.
- RESID. LIFE EST.: PRINCIPAL AIMS OF FEA AND TMF: PREDICTION OF CRITICAL AREAS IN TERMS OF MECH. INTEGR. (PLAST. DEF) + ESTIMATION OF N UNDER ACCEL. TMF TEST
- MAT. CONST. LAW: ISOTROPIC KINEMATIC HARDENING MODEL; MODEL PAR. ST. ISO-T LCF EXP. TESTS ... 3 APPLIED STRAIN RANGES
- FEA: THERMAL TRANSIENT AN. STRUCT. TRANSIENT ANAL.
  ↳ (T COMPARED)
- RLE: MULTI-AXIAL D. MODELS: VON MISES[1], ASME CODE[2], SONSINO-GRUBISIC[3], MANDIL-BROWN-MILLER[4], FATEMI SOCIE[5], SKELTON[6].
- REINFORCEMENT RIBS (CRITICAL FE (POINTS!))

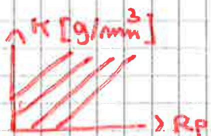
E6

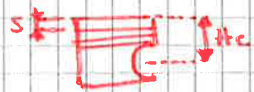
- SKIRT CONTRIBUTES (25%) HEAT FLOW
  1 GAS SEALING RING (GROOVES)
  2 OIL SCRAPER RINGS
- TOP LAND, RING LAND, SKIRT
- HUBS // EDGES OR /\ CONVERGING
- COOLING GALLERY
- MATERIALS: (DIE CAST)
  CAST IRON + { CR-PLATING SUPPORT / MO-PLATING } + Al_2O_3 (ALUMINA) COATING FOR BOWL ADIAB. COND ↑
  Al-ALLOY { PASS. CAR APP. / AlSi9 DIESEL }

SURF. TREAT (GRINDING) → SKIRT, PIN SEATS

- <u>DESIGN GUIDELINES</u>
- SEMI-EMPIRICAL GRAPHS



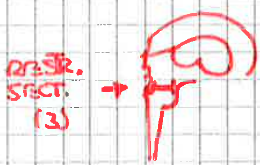




1) D,W → $k = \frac{W}{D^3}$: APPARENT DENS.
$k \rightarrow R_P = \frac{H_c}{D}$: GEOM. RATIO, H_c = COMP. HEIGHT

2) $s = \frac{D}{2}\sqrt{\frac{P_{MAX}}{\sigma_{ADM}}}$: TOP LAND THICK OR $s = f(D)$ FROM TABLES

3) $\sigma_c = \frac{P_{MAX}\, D^2}{D_c^2 - D_i^2} \leq \sigma_{ADM}$ (D_c = GROOVE)



4) $L = \frac{F_{N\,MAX}}{D \cdot P_{ADM}}$: LENGTH OF SKIRT { Al ALLOY 0.98 / CAST IRON 0.44-0.64 } OR $L = 2H_c$

5) GAP PISTON SKIRT-LINER ∝ f(MAT) { Al ALLOY 0.04-0.06 / CAST IRON 0.09-0.11 }

2T

• T14 (DRAWINGS)
- GOOD TH. PROPERTIES: HIGHER LOAD CAPACITY, LOWER SENS. TO MISUSE
- TWO INPUT S, TWO OUT. SH.

• T16 (GRAPHS)
- GEARBOX AUTOMATION REQUIRES
- RADIAL CIRCULAR FLOW
- IF WE DON'T NEED THE EFFECT OF HYDR. COUPLING
- $V_{sh} = C_u$ ENTERING $- C_u$ ENTERED ; W_{out}
- FLOW LOSSES: DISTR. LOSSES ALONG EACH STAGE DUE TO VISCOUS FRICTION
- PRINCIPLE OF SIMILARITY: LINEAR DIMENSION (IN SCALE) $M_1 = \lambda \rho \omega_1^2 D^5$
- $M_E - \Omega_E$; $M_1 - \Omega_{OUT}(\Omega_E)$

T17
- BORN FOR CONFORTABLE FAMILY CARS (AUTOM. GEARBOXES)
- POWERSHIFT: MORE POWER-DISSIPATION DURING GEARSHIFT (TC = POWERS.)
- COUNTERSHAFT POWERSHIFT G. WITH M.D.W.C. (HONDA)
- MULTIDISC CLUTCHES: WE HAVE NOT TO UNLOAD THE IMP. S. DUR. SYNCH. PROC.
- SEMI-AUT F.CLUTCH + T.C. + GEARB. (TC. → STALL TO LOCK-UP)
- $\tau_0 = \frac{\Omega_1}{\Omega_3}\Big|_{\Omega_2=0} = -\frac{z_3}{z_1} = -6 = -2$ $(\tau = \Omega_1/\Omega_2 = 1-\tau_0)$ T18
- IT IS NOT POSSIBLE TO OBT. A GEOM. PROGR.
- $\tau_1 = \frac{z_3}{z_{1dx}}$ $\tau_R = -\frac{z_3}{z_{1sx}}$
- MERCEDES (6, 9): 1 R., 2 S, REAR W.D.C.; ALIGNED I. O. S.
- AUDI (7) 1 R, 1 S, FRONT W.D.C. (ALSO 4WD)

T19 (CVT)
- POSSIBLE SYSTEMS: E. TRANSMISSIONS, POOR TRANSM. EFF.
- VAN DOORN STEEL BELT (HONDA) / LUK CHAIN CVT / R.B.T. CVT
- PITCH RADII OF THE BELT
- BELT: 2 FLEXIBLE STEEL RING BANDS $\tau_{MAX} = 1/\tau_{MIN}$ $\frac{R_{min}}{R_{MAX}} \leq \tau \leq \frac{R_{max}}{R_{min}}$
- FRICTION FORCES BETWEEN PULLEY AND T.E. WOULD NOT WASTE ENERGY IF THERE WERE NO RELATIVE MOTIONS.
- R.B. TOROIDAL CVT: ABLE TO TRANSMIT HIGH TANGENT FORCES WITH LIMITED CONTACT AREA ... THE ADV. HIGH M. EFF. LIMITED SLIP BET. TOROIDAL BODIES!
- INDUSTR. V.: TC: TRANSM. RATIO IS PARTICULAR POOR WHEN ... AND THIS OCCURS WHEN REACTION TORQUE INVERTS ... BLADE ANGLES 2 OPP. FLOW DIR.
SO, THE E. BRAKE EFF. IS DECR. → RETARDER = HIGH DIAMETER WITH LIMITED RADIAL BL. DIM.
→ MODULATE → OIL Q.
- PARTICULAR AUTOMATIC G. FOR URBAN BUSES → RETARDED SPEEDS; TC (= START-UP + RETARDER) 2C, 3B.

POWERTRAIN COMPONENTS DESIGN - EXAM 18/02/21 (S254212)

T

$P_m = R \cdot V = AV_{MAX} + BV^3_{MAX} + CV^5_{MAX} = \eta_t P_{E\,MAX} \rightarrow V_{MAX}$ $\qquad \eta_t = \frac{P_A}{P_E}$

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

$R_A = \frac{1}{2}\rho V_R^2 S C_x \qquad V_R = V \ (hp)$

$R_G = mg\sin(\alpha)$

$\tau_t = \frac{V_{MAX}}{R_R \Omega_E}$

E

$[J_{EQ}]\{\ddot{\Phi}\} + [K]\{\Phi\} = M(\Theta) \qquad [J_{EQ}]$: DIAG. M. $= [Y \times Y]$ $\quad Y = z + 2 \quad [K] =$ TRIDIAG. MATRIX

$J_{EQ} = J_C + m_{CR,R} R^2 + (m_{WP} + m_P + m_{CR,A}) \frac{8 + 2\Lambda^2 + \Lambda^4}{16} R^2 + J_0 \frac{4\Lambda^2 + \Lambda^2}{8}$

$K_{EQ} = \frac{G I_P}{L_{EQ}} \qquad L_{EQ}$ (TUPLIN, CARTER)

$$[K] = \begin{bmatrix} K_E & -K_E & & \\ -K_E & 2K_E & -K_E & \\ & -K_E & 2K_E & -K_E \\ & & -K_E & K_E \end{bmatrix}$$

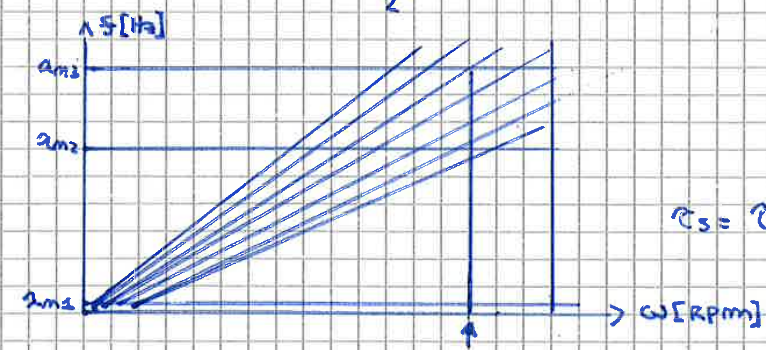
(3 CYL. 4S)

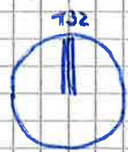
$\delta = ord \cdot \psi$: PHASE SHIFT $\rightarrow$ STAR OF HARMONICS

$ord = \begin{cases} K/2 & \text{4 STROKE E.} \\ K & \text{2 STROKE E.} \end{cases} \qquad \psi = \begin{cases} \frac{4\pi}{z} & 4S \\ \frac{2\pi}{z} & 2S \end{cases}$: ANGULAR PHASE SH. $\rightarrow$ STAR OF CRANKS

$\psi = \frac{4 \cdot 180°}{3}$

MAJOR ORDER : $\frac{3}{2} = 1.5 \qquad \delta = 1.5 \cdot \psi = 1.5 \cdot \frac{4 \cdot 180}{3} = 360°$





$\tau_S = \tau_{S,STAT} + \tau_{S,DYN} = \frac{s A_0}{W_{tS}} + \frac{K_S(\Phi_{i+1} - \Phi_i)}{W_{tS}}$