AIBN analysis report

For: AIBN
UK-000541-DC-004-B

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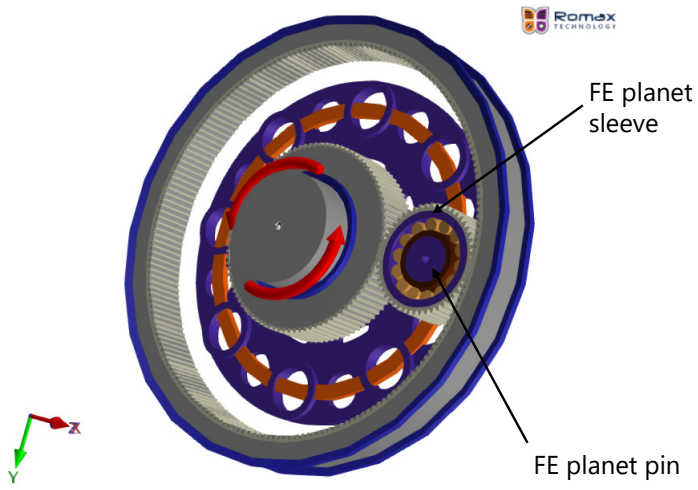
31/05/2018

Model overview

- Two RomaxDesigner models produced, one for each bearing:
  - FAG: UK-000541-RX-002-B
  - SNR: UK-000541-RX-003-B
- Each model contains:
  - Gear set modelled without microgeometry details
  - One planet modelled in detail
    - Solid-meshed FE planet pin and sleeve for single planet
    - Rolling element contact based on thin-strip model and Hertzian contact theory
  - Planet carrier modelled as FE. Stiffness of FE set very high to remove deflections of the planet carrier.



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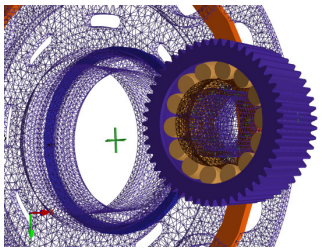
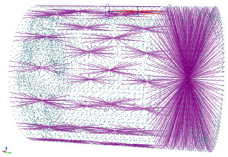
Bearing modelling

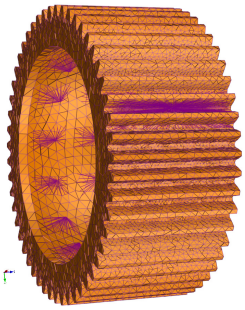
- Bearing outer race integral with planet sleeve (solid-meshed FE component) – permits out-of-round deformation of gear
- Inner ring, planet pin, carrier modelled as solid-meshed FEs, permits out-of-round deformation
- Gear contact modelled as a connection distributed with rigid (RBE2) elements across face width. This adds some rigidity but does not prevent out-of-round deformation.
- Nonlinear local contact stiffness/deflection calculated by Romax's thin-strip model accounting for internal geometry, deflections and misalignment
- Calculated roller-raceway loads distributed to raceway with load-distribution (RBE3) elements
- (Separate *Romax**DESIGNER*** model for FAG and SNR bearings for different bearing geometry)
- Planet carrier stiffness increased to remove planet carrier deflections. This is required as the other 7 planets are not included in the model



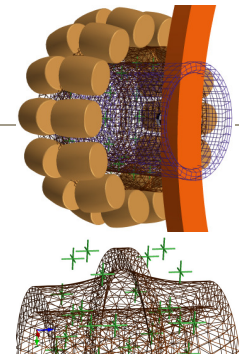
FE meshes for pin, carrier, inner race and gear



Pin is a solid-mesh. RBE2 spiders connect to the carrier to a single point, and in two rings of nodes to the bearing inner race.



Gear has RBE2 connection to mesh point and RBE3 connection for bearing rollers



Bearing rollers connected to raceway via RBE3 spiders (not shown) from nodes on roller centre

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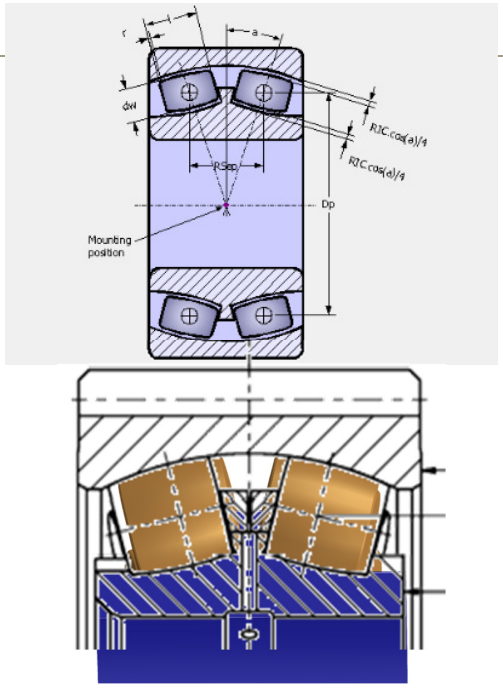
Bearing modelling: Internal details

- Bearing rollers and Race crowning modelled as symmetric about their centre.
  - *This is due to current software UI limitations*
  - *Effect on raceway stresses expected to be small as correct race and roller crowning values have been applied.*
- Roller corner chord along roller axis modelled as the same on both ends of the roller.
  - This is correct for the SNR bearing
  - The FAG bearing has different lengths on either side of the roller, (contact does not approach the ends of the roller in the cases analysed)
- Romax model and actual bearing have same Pitch Circle and Roller Diameter and same distance between rows at pitch diameter
- Romax model has length of rollers symmetrical about "pitch point"



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Dynamic/static capacity and material properties

- To do back to back comparison capacities were calculated using bearing geometry
  - Dynamic capacities calculated according to ISO 281, static as per ISO 76
  - The ISO standard *does not account for the difference in curvature in the two bearings* so the capacities are very similar. If the curvatures were accounted for it would be expected that the SNR bearing would have a capacity greater than the FAG.
  - The ISO standard does not account for any difference in material properties – it is assumed typical bearing steel properties

Bearing	FAG	SNR
Dynamic capacity (kN)	243.8	240.4
Static capacity (kN)	289.4	285.1

- Two material used in model
  - 16NDC13: Planet pin/sleeve, Ring gear, sun gear
  - M50: bearing inner race, bearing roller

Material	16NCD13	M50
Young's modulus (GPa)	197	202
Density (kg/m^3)	7850	7800
Poisson's ratio	0.3	0.29
Thermal expansion coefficient (µm/m/°C)	11.5	11.6

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Bearing clearance

- Operating bearing clearance is calculated using hoop stress calculations based on initial (off-the-shelf) clearance, mid-tolerance inner race fit and assigned operating temperatures
- Lowest tolerance value taken as initial clearance
- Fits applied (interference fit on shaft) yields reduction in clearance
- Operating temperature assumed to be 100°C for all bearing components and shafts
  - Initial temperature of all components assumed to be 20°C
  - As all parts have similar thermal expansion coefficient the change in fit due to temperature is not significant

Bearing	FAG	SNR
Initial clearance (µm)	117	120
Clearance after temperature and fits applied (µm)	110.61	114.15

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Modelling assumptions and load cases

- No power losses calculated (power in = power out)
- Lubricant assumptions for a_{ISO} operating conditions modification factor calculations
  - Filtered to ISO 4406 contamination code: 15/12
  - Includes EP additives
  - Operating temperature: 100°C, Kinematic Viscosity at 100 °C assumed 11.6mm^2/s (based on Total Aerogear 1032)
- Gravitational force not included (not significant relative to applied loads)
- Centrifugal forces not included (not significant relative to applied loads for this low speed system)
- Assumed equal load share between the planets, reduced the torque by 1/8 as only one planet in the model.
- Stress results extracted for 4 load cases:
  - Values are defied at the output (rotor)

Load case	Power (kW)	Speed (rpm)	Torque used in Romax model [1/8 actual torque] (Nm)	Torque (Nm)
TOPtrans (EC225LP)	2512	275	10904	87227
MCP (EC225LP)	1959	265	8824	70593
TOPtrans (AS332L2)	2250	265	10134	81079
MCP (AS334L2)	1768	265	7964	63710

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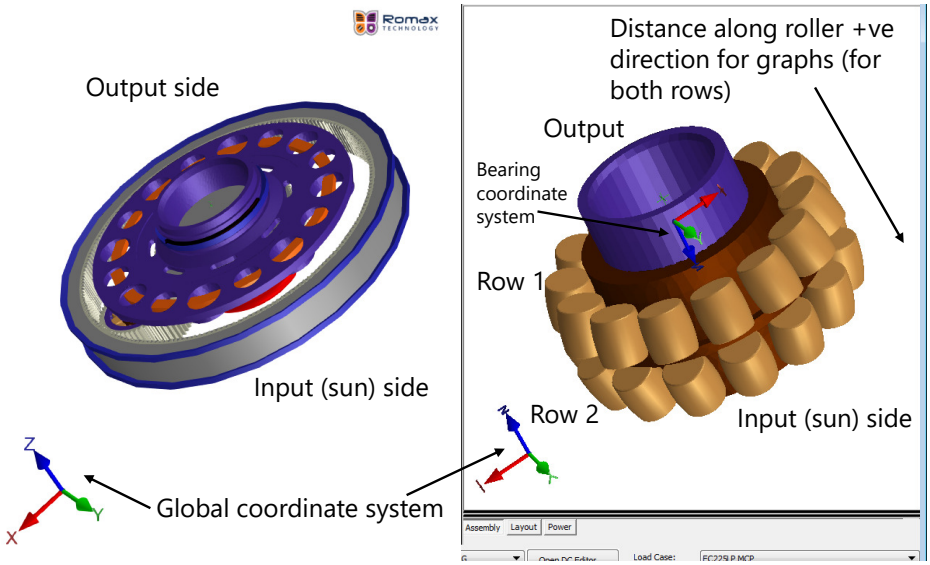
RESULTS OVERVIEW

Results coordinate systems

- Row 1 is top (output) side
- Row 2 is bottom (input) side
- Direction along rollers increasing in direction towards ground



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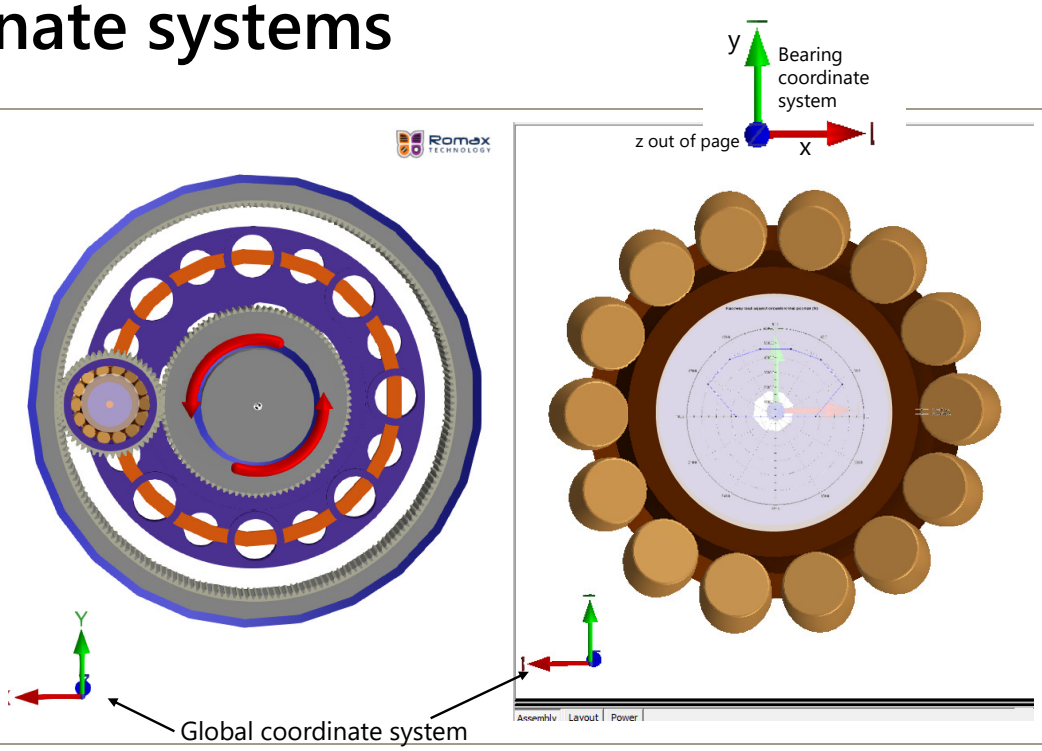
Results coordinate systems

- Red arrow shows direction of rotation and input torque on sun
- Bearing polar plots are shown relative to bearing local coordinate system as illustrated



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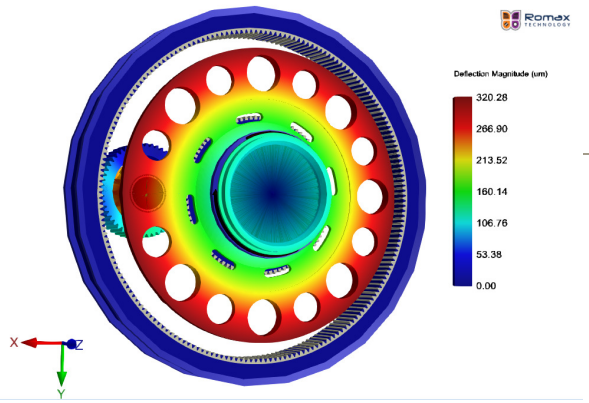
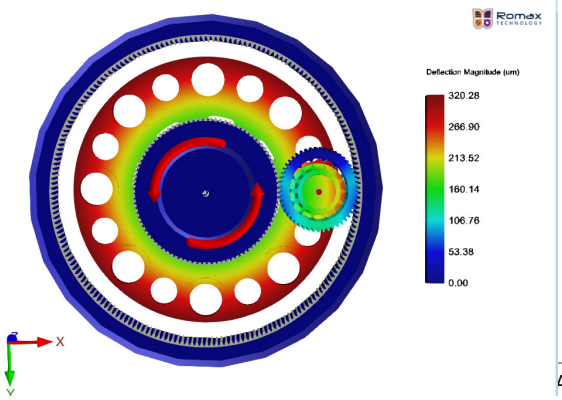
System deflections

- System deflections from:
  - UK-000541-RX-002-B (EC225LP MCP)
- As only one planet is included in the model planet carrier deflections will not be representative. As such the carrier has been stiffened so that it deflects minimally under the applied loads.





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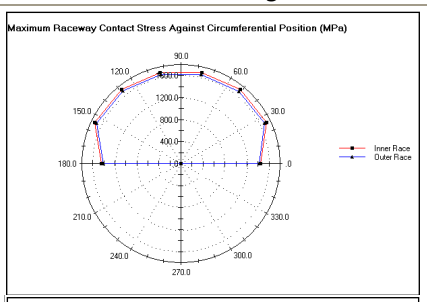
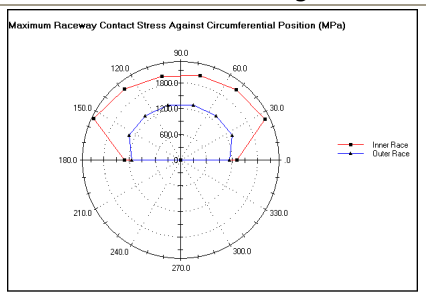
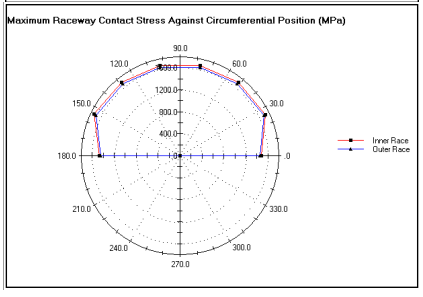
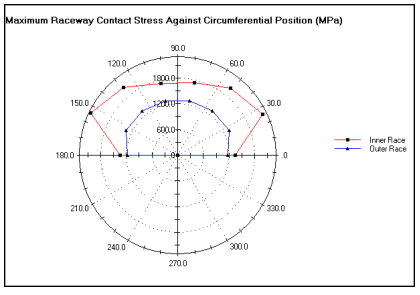
Element load distribution (EC225LP MCP)

- Linear coordinate gives the value of race way stress
- Polar coordinate gives the angular position around the bearing.
- Y-axis is in direction of load on bearing
- The black points denote the position of the rollers (results are shown for a single position)
- Both FAG and SNR bearings show similar load distributions on their rollers.
- SNR bearing outer race maximum contact stress is lower than that of the FAG bearing.
- The SNR bearing shows very high inner race contact stresses. This is due to the roller contacting the inner race undercut corner and is investigated later.
- Note that peak stress does not appear at the central loaded rollers but those at the edges.
  - This shows that the bearing rings have deformation that is affecting the load distribution

	FAG Bearing	SNR Bearing
Row 1		
Row 2		

Note : graphs have different scales

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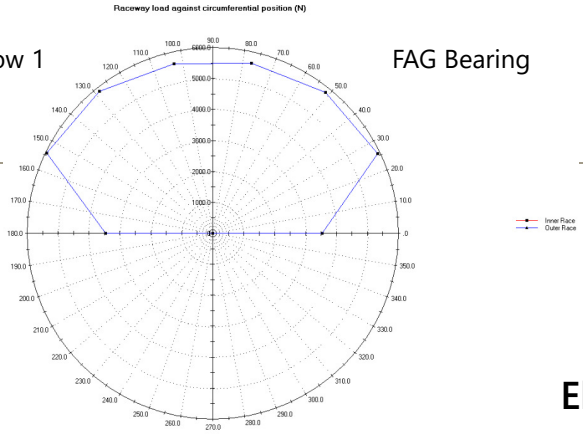
Row 1

FAG Bearing



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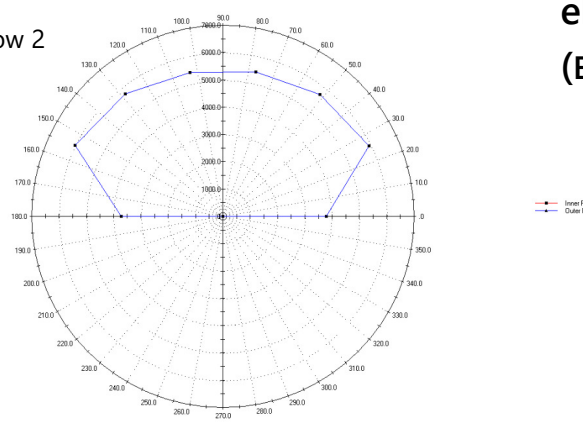
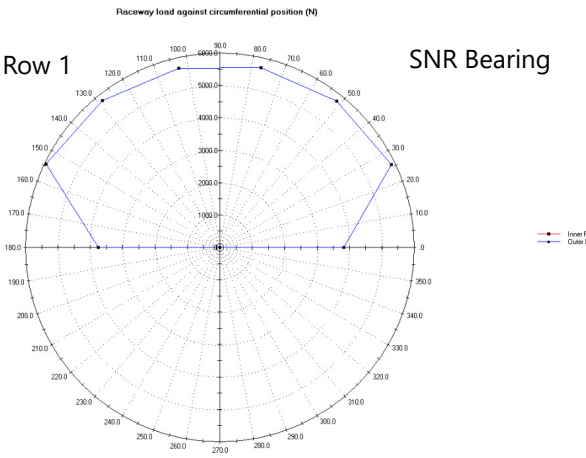


Element load vs. element angle (EC225LP MCP)

Row 1

SNR Bearing



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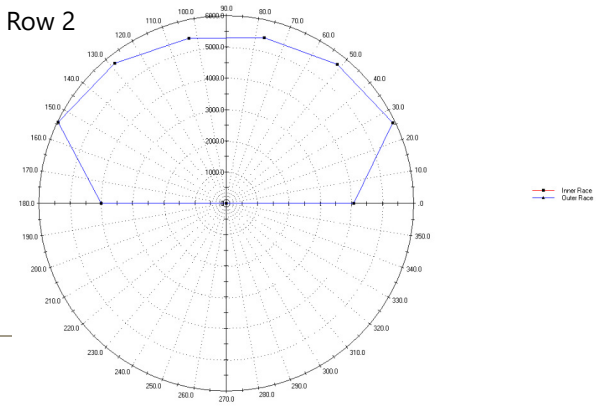

Slide 13

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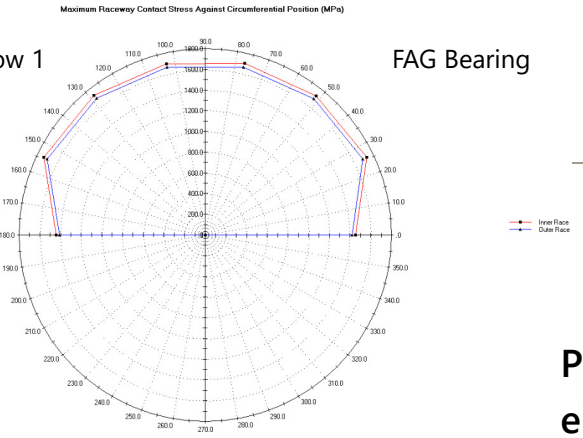
Row 1

FAG Bearing



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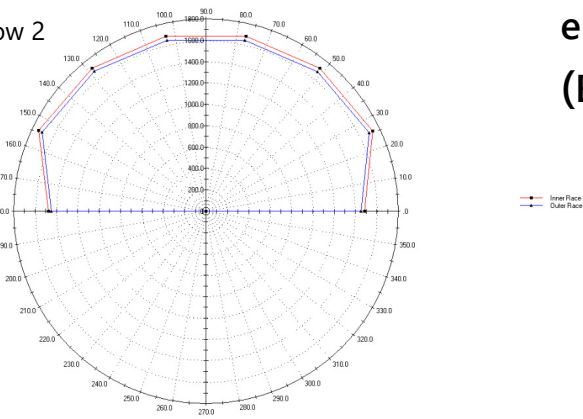
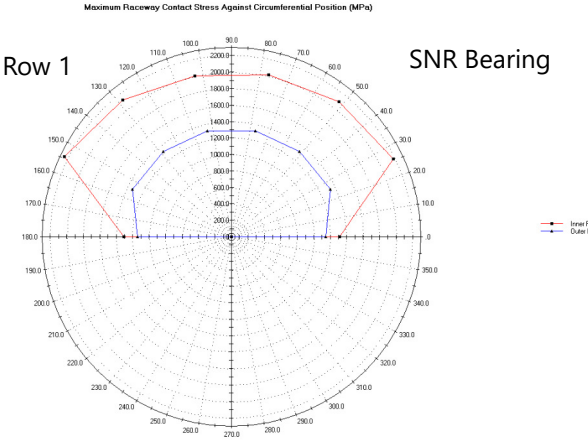


Peak stress per element vs. element angle (EC225LP MCP)

Row 1

SNR Bearing



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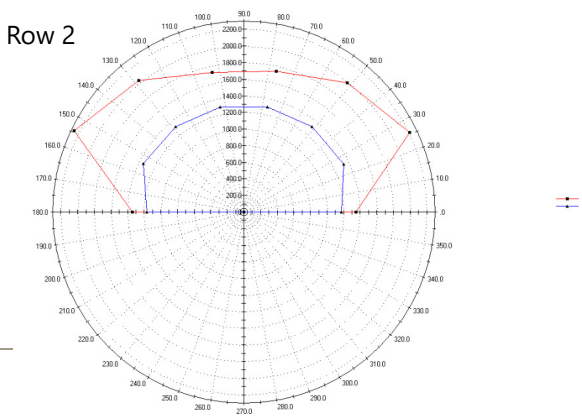
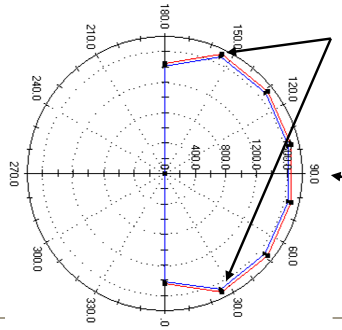
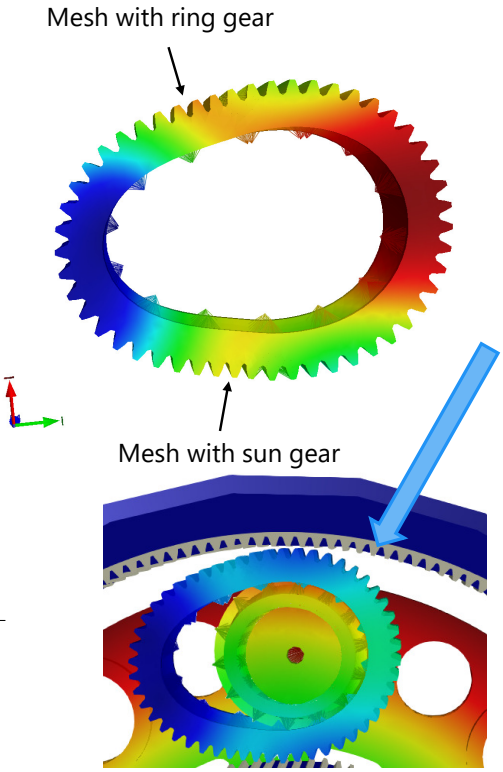


Slide 14

Gear sleeve deflection (EC225LP MCP)

- The image shows the gear rim flexing under load (deflection exaggerated)
- The sides of the gear appear to bulge.
  - The top and bottom of the gear moves inwards
- The deformation of the gear blank/bearing outer race explains the load distribution seen.
  - Load distribution plot orientated to match gear





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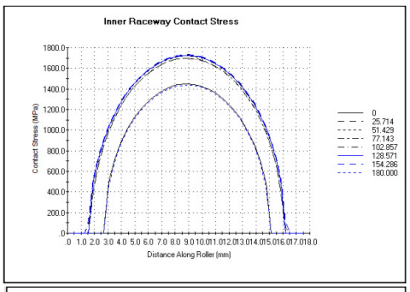
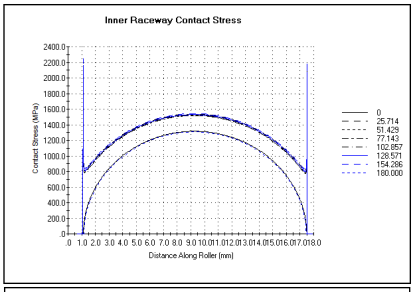
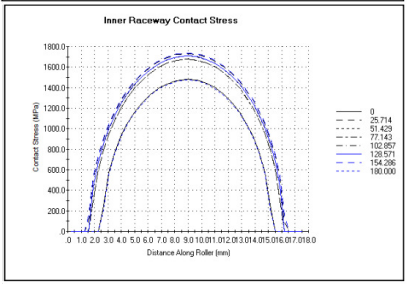
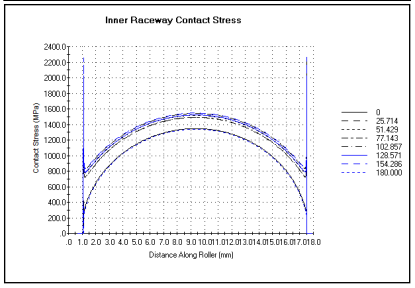
Inner Race contact stress (EC225LP MCP)

- Images show stress distribution along loaded rollers.
- FAG bearing shows more centralised contact than the SNR bearing
- The central peak stress of the FAG bearing is greater than the SNR bearing.
- The SNR bearing shows spikes of stress at the edges of the roller (see later slides).
- The FAG bearing contact ellipses do not extend to the edges of the contact zone, due to the greater relative radius of curvature

	FAG Bearing	SNR Bearing
Row 1		
Row 2		

Note : graphs have different scales

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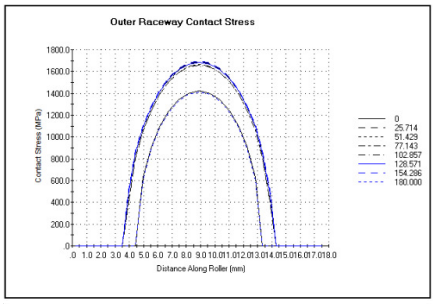
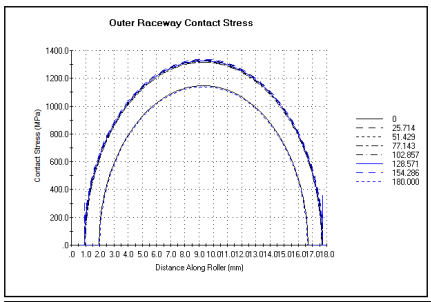
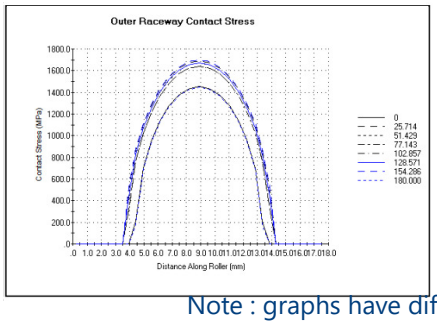
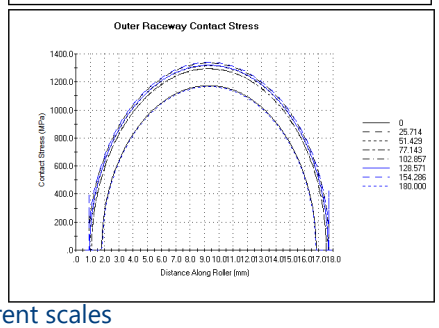
Outer race contact stress (EC225LP MCP)

- SNR bearing shows a lower central peak contact stress than the FAG bearing.
- The SNR Roller is in contact over more of its length. The FAG bearing contact region is much more compact
- In this load case case there is no edge stress in the SNR bearing.

	FAG Bearing	SNR Bearing
Row 1		
Row 2		

Note : graphs have different scales

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Peak contact stress (EC225LP MCP)

- SNR Bearing shows lower inner and outer race (results below for EC225LP MCP)
  - SNR bearing edge stresses shown in brackets

Bearing	FAG		SNR	
Row	1	2	1	2
Max. inner race stress (MPa)	1734	1739	1543 (2249)	1546 (2266)
Max. outer race stress (MPa)	1696	1700	1337 (N/A)	1339 (N/A)

- The lower stress in the SNR bearing is due to closer conformity between the raceway and roller.
- For this load case the SNR inner race edge stresses are greater than the central contact stresses. The results given in the table above are the peak central contact stresses.

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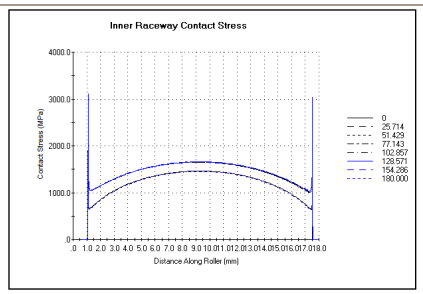
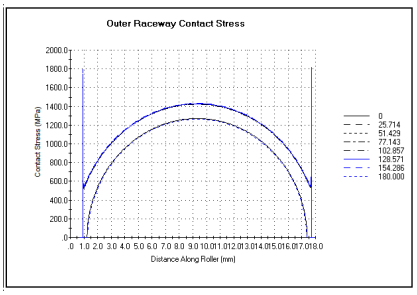
SNR bearing edge loading – (EC225LP TOPtrans)

- Results are shown for the largest torque case considered
- SNR bearing shows edge loading on both the inner and outer race
  - This is more prominent in higher torque load cases
- This edge loading is caused by the close conformity of the inner and outer races to the roller.
  - The corners of the FAG bearings never come into contact.
  - The inner race of the SNR bearing has greater conformity as such it shows edge contact under the same load.





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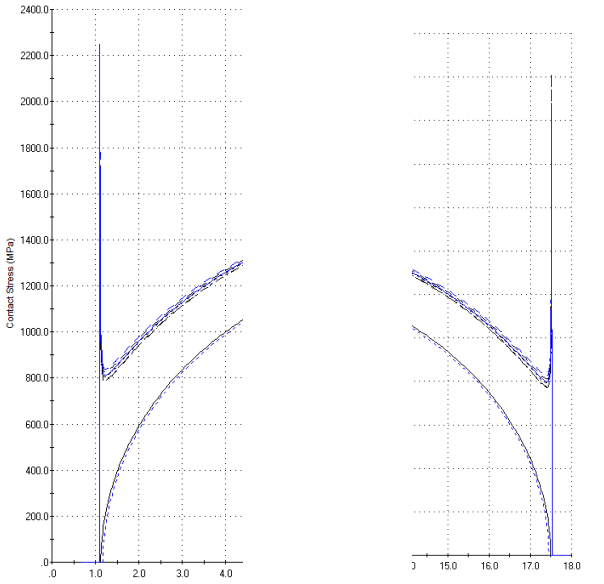
Edge loading

- We have introduced the undercut at 1.1mm in from the roller edge.
- We have refined the roller discretization to capture more stress points, but we still only have a few in the area of the relief.
- The actual profile around this start of undercut point is not controlled in manufacture (we believe), thus the stress predictions around this point have considerable uncertainty.
- If there is a discontinuity in curvature – either in slope on the raceway or at the roller corner – there will be a level of stress concentration.
- High stress around this point might result in wear that would relieve this stress – it does not necessarily mean it would progress to a problem or even visible wear.
- However, this calculated stress is used in the life calculations



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a_{iso} factor (life modification factor)

- a_{iso} is a adjustment factor that takes the basic L_{10} stress and modifies it to account for the operating condition. a_{iso} considers the effect of:
  - Contamination grade, e_c
  - Viscosity ratio,
  - Fatigue load limit C_u
- C_U used in our ISO 281 rating is calculated using ISO 281-2007 B.3.3.3 ("simplified" method)
  - This method is only based on the static capacity and ball pitch circle diameter, so does not account for raceway curvature
- C_U used in our ISO T/S 16281 rating is calculated using B.3.2.2.4 ("advanced method")
  - This method accounts for the curvature of the race and roller as such there are large differences between the SNR and FAG bearings
  - aiso,r is a weighted average – a factor is calculated for each strip in the contact analysis
  - aiso,r would be higher in the SNR bearing in both load cases due to the more conformal geometry/lower stress in the centre of the contact patch however, the presence of the high edge stress at TOPtrans pushes the value down.

	FAG		SNR	
Load case	a_{iso} (ISO 281)	$a_{iso,r}$ (ISO TS/16281)	a_{iso} (ISO 281)	$a_{iso,r}$ (ISO TS/16281)
EC225LP MCP	0.91	0.50	0.90	0.80
EC225LP TOPtrans	0.75	0.45	0.74	0.48
AS332L2 MCP	1.01	0.52	1.00	0.87
AS332L2 TOPtrans	0.80	0.46	0.79	0.56

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Bearing lives: ISO 281 & ISO T/S 16281

- ISO 281 results predicts similar FAG lives to SNR lives, as the capacity, a_{iso} factor and equivalent load are all very close
- ISO TS/16281 results predict greater lives for the SNR bearing in the lower load situations, due to the radius of curvature difference
- In the higher load situations, the contact stress at the edge becomes more significant, and thus the difference is life is not as substantial.

	FAG		SNR	
Load case	IS0 281 life – L10m (hrs) [C_U simplified method]	ISO TS/16281 life – L10mr (hrs) [C_U advanced method]	IS0 281 life – L10m (hrs) [C_U simplified method]	ISO TS/16281 life – L10mr(hrs) [C_U advanced method]
EC225LP MCP	3280	2256	3086	7823
EC225LP TOPtrans	1277	1044	1203	1540
AS332L2 MCP	5130	3247	4822	11789
AS332L2 TOPtrans	1809	1377	1703	2695

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Life calculation intermediate values, EC225LP TOPtrans Case

- FAG

	Raceway Dynamic Load Rating (N)		Raceway Dynamic Equivalent Load (N)		Reference Life L_{10r}		Dynamic Equivalent Reference Load (N)	Reference Life L10r (MRevs)		Reference Life L10r (hrs)	
	Inner Q_{ci}	Outer Q_{ce}	Inner Q_{ei}	Outer Q_{ee}	(MRevs)	(hrs)	P_{ref}	Inner (Qci/Qei)^4	Outer (Qce/Qee)^4	Inner (Qci/Qei)^4	Outer (Qce/Qee)^4
Row 1	34228.1	57067.8	7994.3	10187.4	266.451	4223.4618		336.05	984.72	5329.12	15615.54
Row 2	34228.1	57067.8	7921.5	10101.5	276.209	4378.1325		348.58	1018.64	5527.74	16153.52
Bearing					146.476	2321.761	54623				

- SNR

	Raceway Dynamic Load Rating (N)		Raceway Dynamic Equivalent Load (N)		Reference Life L_{10r}		Dynamic Equivalent Reference Load (N)	Reference Life L10r (MRevs)		Reference Life L10r (hrs)	
	Inner Q_{ci}	Outer Q_{ce}	Inner Q_{ei}	Outer Q_{ee}	(MRevs)	(hrs)	P_{ref}	Inner (Qci/Qei)^4	Outer (Qce/Qee)^4	Inner (Qci/Qei)^4	Outer (Qce/Qee)^4
Row 1	33754.6	56070.6	7623	7088.6	360.952	5721.3834		384.44	3914.70	6096.41	62079.04
Row 2	33754.6	56070.6	7483.9	7034	387.399	6140.5922		413.83	4037.67	6562.45	64029.10
Bearing					201.798	3198.6606	48915.5				

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Life calculation intermediate values, EC225LP MCP case

- FAG

	Raceway Dynamic Load Rating (N)		Raceway Dynamic Equivalent Load (N)		Reference Life L_{10r}		Dynamic Equivalent Reference Load (N)	Reference Life L10r (MRevs)		Reference Life L10r (hrs)	
	Inner Q_{ci}	Outer Q_{ce}	Inner Q_{ei}	Outer Q_{ee}	(MRevs)	(hrs)	P_{ref}	Inner (Qci/Qei)^4	Outer (Qce/Qee)^4	Inner (Qci/Qei)^4	Outer (Qce/Qee)^4
Row 1	34228.10	57067.80	6810.20	8703.60	504.58	8299.86		638.10	1848.28	10498.59	30409.42
Row 2	34228.10	57067.80	6754.60	8639.80	520.99	8569.81		659.38	1903.49	10848.56	31317.64
Bearing					276.85	4553.82	45126.70				

- SNR

	Raceway Dynamic Load Rating (N)		Raceway Dynamic Equivalent Load (N)		Reference Life L_{10r}		Dynamic Equivalent Reference Load (N)	Reference Life L10r (MRevs)		Reference Life L10r (hrs)	
	Inner Q_{ci}	Outer Q_{ce}	Inner Q_{ei}	Outer Q_{ee}	(MRevs)	(hrs)	P_{ref}	Inner (Qci/Qei)^4	Outer (Qce/Qee)^4	Inner (Qci/Qei)^4	Outer (Qce/Qee)^4
Row 1	33754.60	56070.60	5742.70	5983.60	1077.02	17715.82		1193.62	7710.64	19638.42	126861.52
Row 2	33754.60	56070.60	5685.70	5941.80	1119.43	18413.47		1242.21	7929.92	20437.86	130469.20
Bearing					592.84	9751.58	35403.00				

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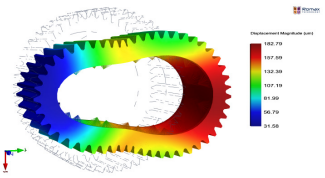
Conclusions

- The planet carrier has been included in the model, however due to only a single planet gear being modelled the stiffness of the carrier has been set such that it deflects minimally under the applied loading. If representative material properties are used the carrier deflections are unrealistically high.
- The planet gear deforms under load and affects the load distribution at the rollers. The load is more evenly shared over the rollers than would be expected with a more rigid outer race. The loads on the rollers are very similar with peak load appearing at the "sides" of the gear where the meshes are compressing the gear. A rigid gear would be expected to result in the highest loaded roller being the load direction with a greater magnitude of contact stress than with the flexible outer gear. The deformation of this gear as it rotates will introduce stress cycles and affect the stress that results due to bending in the roots of the gear teeth. (This is the same for both bearings).
- The SNR bearing has lower contact stress across the majority of the roller due to the closer conformity of the roller and raceway. However, this closer conformity leads to the contact footprint progressing very close to the edge of the roller. This means the edge of the raceway where the undercut starts will come into contact. At this point there will be a stress concentration. As this is not (as far as we know) a carefully controlled radius our predicted stress here is subject to some uncertainty.
- The ISO 281: 2007 life calculations are presented but these do not take into account the difference in relative curvature so the calculations for the FAG and SNR are very similar.
- The ISO T/S 16281 calculations account for the high edge stresses and use the advanced methods for calculating the fatigue load limit that is used in the a_{iso} calculation. In lighter load cases where the edge load is not significant the life of the SNR bearing is much greater than the FAG bearing. In heavier load cases the SNR life is still reported as greater than the FAG life, but only very slightly as the high edge stresses we are predicting are reducing the life. Note that in practice the edge stresses may be absent, current experience has not highlighted any issues in this region.



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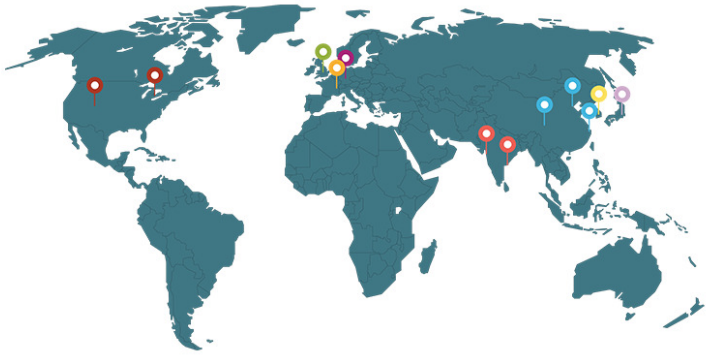

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