

GENEVA MECHANISMS. PART 4: ADVANCED EXAMPLES

1. NOTATION AND UNITS

Only those symbols directly used within this Item are given. Consult Data Item 96002 (Reference 2) for the full notation and Units for Geneva mechanisms. The units used in this Item are predominantly SI. All angles are measured in an anti-clockwise sense and the crank rotates anti-clockwise.

		<i>Units</i>
T	duration of (start-up or shut-down) motion phase	s
t	time	s
t_i	time after start of motion segment at which crank angle ϕ_i occurs	s
α	angular acceleration of crank during start-up or shut-down	rad/s ²
ϕ_i	crank angle at increment i	rad
ϕ_0	initial crank angle (at beginning of start-up or shut-down)	rad
$\dot{\phi}_i$	angular velocity of crank at incremental angle i during transient motion	rad/s
$\dot{\phi}_0$	angular velocity of crank before transient motion segment starts	rad/s
ω	nominally constant angular velocity of crank	rad/s

2. INTRODUCTION

The examples in this Item provide insight into the application of Data Item 96002 (Reference 2) for advanced design analysis of Geneva mechanisms. Data Item 96035 (Reference 4) presents a series of examples of more basic Geneva mechanism design problems and describes in detail the use of ESDUpac A9611 software for their solution. Aspects of the design process treated here include determination of the following:-

- (i) geometric relationships between the wheel, crank and roller (or pin),
 - (ii) the wheel motion,
 - (iii) the forces and torques,
 - (iv) the contact stress,
 - (v) the lubrication film thickness and regime (if applicable)
- and (vi) the roller performance (if applicable).

Data Item ME3 (Reference 1) presents the underlying theory used to analyse a Geneva mechanism and identifies the interdependent relationships between its design parameters. Analysis of the mechanism is carried out using the ESDUpac A9611 software, described in Data Item 96011 (Reference 3).

Example 1 (Section 3) demonstrates the analysis of an external Geneva mechanism in an existing machine in which the crank angular velocity varies during each rotation. The performance of the mechanism is analysed when the nominally constant crank angular velocity is raised from 180 to 250 rev/min. The external torque driven by the mechanism cannot be adequately represented by one of the (four) standard models provided in the ESDUpac A9611 software so a user-defined external data file is used. An analysis of the mechanism using ESDUpac A9611 and external motion and torque data files created by the user is described. The results of the analysis indicate that the contact stress exceeds the stated limit. A method to identify the dominant factor causing the high contact force is explained and various methods of eliminating operational problems caused by excessive contact stress are investigated and compared.

Example 2 (Section 4) describes how transient effects that occur during start-up or shut-down of an external Geneva mechanism can be analysed. The mathematical approach for developing a user-defined crank motion data file to represent the acceleration of the mechanism from rest to operating speed (and vice-versa during shut-down) is described (see Appendices B and C). Two alternative start-up times, i.e. the time taken to reach full operating speed, and two corresponding shut-down times, are analysed for an initial crank angle of zero. With the information gained from these analyses, the initial crank angles that would result in the highest drive torque requirement for the two start-up/shut-down times, are found. These 'worst cases' are analysed and recommendations made for the control of the mechanism during start-up.

3. EXAMPLE 1. EXTERNAL GENEVA MECHANISM WITH VARYING CRANK ANGULAR VELOCITY DRIVING A WHEEL WITH A COMPLEX VARIATION IN TORQUE LOADING

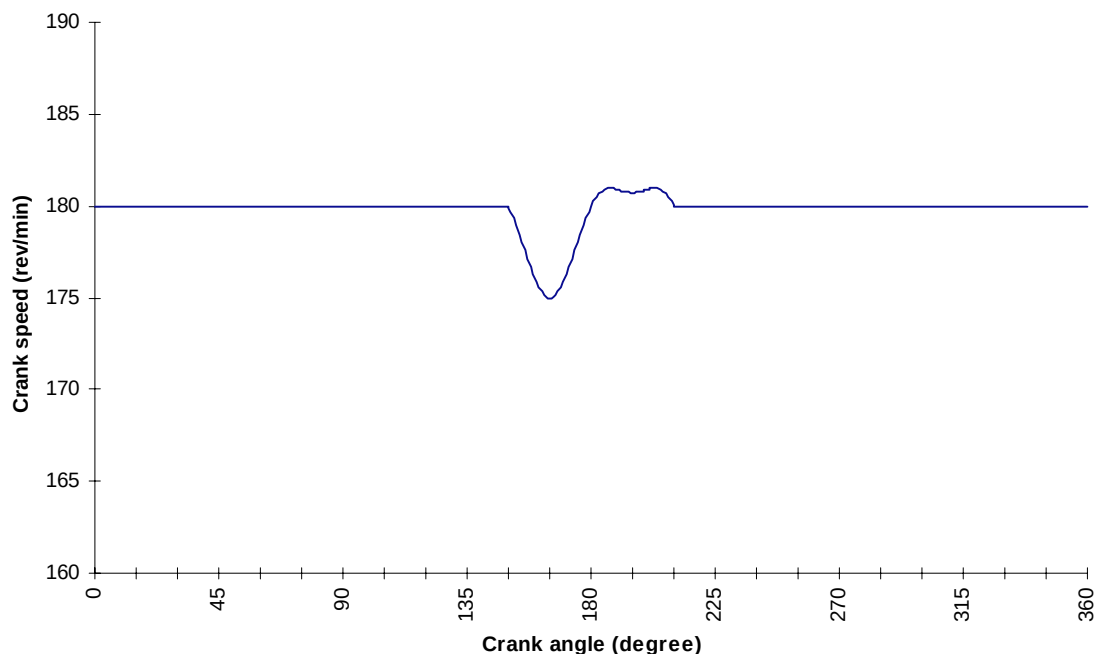
3.1 Problem Statement

A machine that drives a linkage which moves components into position for processing incorporates an intermittent motion mechanism. The linkage is stationary while processing takes place and moves components in and out of position when in motion. The machine operates in 'average working conditions' with negligible vibration forces from external sources.

A six-slot external Geneva mechanism operates satisfactorily when driven by a 1.5 kW geared-motor running at a nominally constant angular velocity of 180 rev/min. To increase productivity, the crank speed is to be increased to 250 rev/min.

The actual motion of the crank and the external torque due to the resistance to motion of the shaft driven by the Geneva during each crank rotation have been measured experimentally. The crank speed, measured for each degree of crank angle for the nominal crank speed of 180 rev/min, is plotted in Sketch 3.1. The method for processing the experimentally-measured torque values is discussed in Section 3.3.

Sketch 3.1 shows that the crank speed varies during the indexing phase of the motion. A speed drop occurs before the control system of the motor can compensate for the increasing torque as the crank roller bearing enters the slot and the wheel accelerates. Next a small over-speed occurs as the torque starts to decrease after the crank angle reaches 180° and the roller begins to move out of the slot.



**Sketch 3.1 Measurement of crank speed
(nominal value 180 rev/min)**

3.2 Preliminary Design Decisions

The Geneva wheel has an outside diameter of 250 mm. It is attached to its shaft by a 48 mm outside diameter bolted flange. The locking radius of the wheel is 40 mm and the six slots are 30 mm wide and 70 mm long. The slot faces are hardened and ground to a maximum surface roughness of 1.5 microns from steel with Young's modulus and Poisson's ratio of 205 GN/m² and 0.29, respectively, and the allowable contact stress is limited to 850 N/mm².

Lubrication is provided by oil with a pressure exponent and dynamic viscosity of $18 \times 10^{-9} \text{ m}^2/\text{N}$ and 0.013 N s/m^2 , respectively.

The total mass and mass moment of inertia of the rotating parts of the wheel and its drive shaft are 3.5 kg and 0.08 kg m^2 , respectively. The mass centre of the wheel coincides with the centre of the wheel shaft bearing.

The total mass and mass moment of inertia of the crank, its flywheel and the rotating parts of the drive assembly are 1.0 kg and 0.002 kg m^2 , respectively. Although the mass centre of the crank itself is offset 9 mm from the crankshaft bearing, a counter balance weight is incorporated in the flywheel to make the mass centre of the crank assembly coincident with the crank shaft bearing. This procedure is carried out using the method described in Data Item 89028 (Reference 8). Note that the size chosen for the flywheel is a compromise between having sufficiently large inertial properties for the speed fluctuation of the mechanism to be minimised for a given load and avoiding an excessive acceleration torque requirement during start-up. Example 2 of this Data Item considers the significance of the acceleration torque during start-up (see Section 4).

A 30 mm diameter roller bearing is selected from the list in Appendix A. It must operate for at least $50\,000$ hours with a reliability of 90% . Both the bearing width and the width of the wheel slot faces are 14 mm .

Neither a special material nor a sophisticated oil feed system are specified for the bearing and both the material and lubrication factors are taken as 1.0 , based on guidance in Section 4.6 of Data Item 96002 (Reference 2). Based on guidance in Table 4.5 in Data Item 96002, the service factor is taken as 1.5 because the mechanism operates under 'normal conditions' but the load may periodically fluctuate above the experimental values. From guidance in Table 8.5 of Data Item 81005 (Reference 7), the static safety factor is taken as 1.0 since the mechanism operates in average working conditions with 'normal demands' on quiet running.

3.3 Modelling External Torque Using an External Torque Data File

It is usually quicker to develop a spreadsheet rather than write a specific computer program to process the measured data. The procedure for converting the data to the format specified for the external torque data file to be imported into ESDUpac A9611 is described in Appendix B of Data Item 96011 (Reference 3). A copy of the comma-delimited format file created in this example (named GENLOAD.CSV) is installed in the working directory of ESDUpac A9611 for versions later than 1.1 of the program.

3.4 Analysis of the Mechanism at 180 rev/min

In what follows the analyses are named $9901\text{EX}1^*$, where the asterisk is replaced by a letter A, B, C *etc.* to indicate the case under consideration, and these names are also used to define the files involved in running the program, following the convention adopted in Section 3.3.7 of Data Item 96035 (Reference 4). The initial analysis is named $9901\text{EX}1\text{A}$ and is made to assess the behaviour of the mechanism for the nominally constant crank speed of 180 rev/min using ESDUpac A9611 with the data described in Sections 3.2 and 3.3. The results of the analysis, summarised in the descriptive output file $9901\text{EX}1\text{A}.\text{OUT}$ produced by the program, includes two warnings.

3.4.1 Consideration of warnings given

In the data input section of file $9901\text{EX}1\text{A}.\text{OUT}$, the program warns "Non-zero external torque specified at the point of engagement". The user is cautioned to ensure that this torque is not actively rotating the wheel as the crank roller enters the slot. Note that the software can only issue this warning when an external torque data file is used. In this example, the external torque data represent the resistance to motion of the

output shaft and not an applied torque that would rotate the wheel. The warning can therefore be ignored.

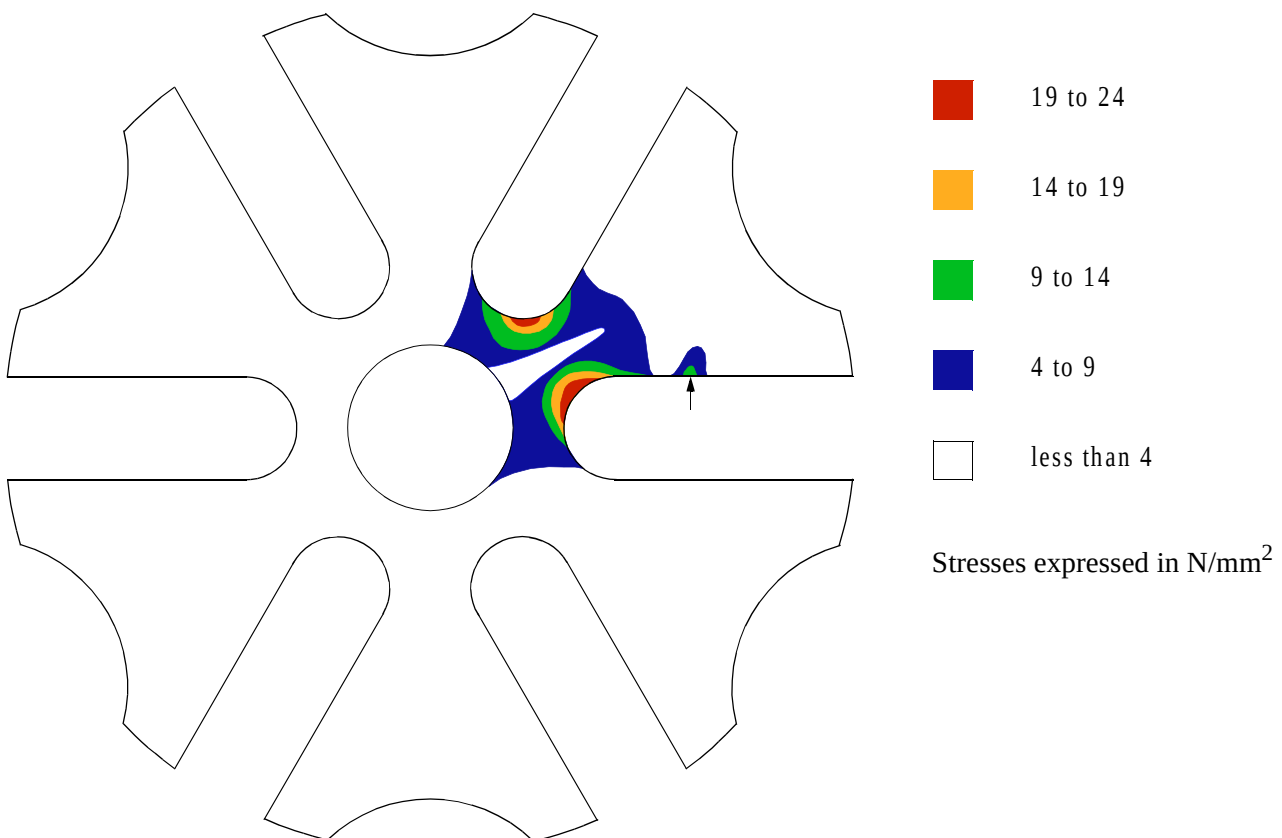
At the end of the data output section, a warning is given that the distance between the slot root and the hub is less than the roller/pin diameter. In the 'Mechanism geometry' section of the output data in the descriptive output file, the gap between the slot roots and the 48 mm diameter 'hub' is given as 16 mm. A recommendation is given to check that the component has adequate strength in this region and that recommendation is followed using a finite element analysis.

3.4.2 Assessing the possibility of component failure using finite element analysis

A finite element analysis (FEA) of the stresses and deformations in the Geneva wheel resulting from contact between the roller bearing and the wheel slot surface is carried out. As recommended in Section 5.3.5 of Data Item 96002 (Reference 2), the contact force at the value of maximum torque is applied as a point force.

This force is applied to the FEA model at the node closest to the position specified in the descriptive output file produced by ESDUpac A9611. The contact force at maximum torque is given in the descriptive output file as 736.1 N. This force is applied normal to the wheel slot contact surface 76 mm radially out from the wheel shaft centre. The arrow shown on Sketch 3.2 illustrates the location of the point of application of this force and its direction.

The von Mises stress distribution plot produced by the FEA is shown in Sketch 3.2.



Sketch 3.2 Von Mises stress distribution in Geneva wheel

The maximum von Mises stress in the wheel is predicted to be 23.7 N/mm². This stress occurs on the surface of the bottom of the slot, as shown in Sketch 3.2. A linear elastic high-cycle fatigue analysis would show that structural failure of the wheel is highly unlikely for the calculated stresses.

3.4.3 Geneva mechanism performance at 180 rev/min

Results of calculations listed in the 'Mechanism geometry' section of the descriptive output file confirm the crank length and distance between shaft centres of 72.17 and 144.3 mm, respectively.

The maximum contact stress of 821 N/mm² is within the allowable limit for the material of 850 N/mm².

The roller life is predicted as infinite (*i.e.* greater than 10⁶ hours) and output from the program therefore warns that the design data should be reviewed, in case the data specification of very low loads and/or speeds is not representative of the actual design. In this example, the prediction of infinite life implies the absence of bearing failures during the required operating life of the machine at 180 rev/min. The 100% boundary lubrication regime is undesirable but typical of the contact between a crowned roller bearing and flat wheel slot face.

The peak drive torque and power requirement are predicted as 31.5 N m and 593.7 W, respectively. The power requirement is therefore well within the 1.5 kW capacity of the motor.

The theoretical (see Section 5.6 of Data Item 96002 (Reference 2)) limiting speed for the mechanism is 190 rev/min beyond which the contact stress limit is exceeded.

3.5 Analysis of the Mechanism at 250 rev/min

All files for the analysis with the crank speed set to 250 rev/min are named 9901EX1B. Output from the program confirms that all dimensions are unchanged from analysis 9901EX1A. (They are independent of the crank speed.) The maximum power requirement increases by approximately 2½ times from 593.7 to 1505 W for the speed increase of approximately 40%, while the maximum torque rises by approximately 80% from 31.5 to 57.5 N m.

Although the modified life rating of the roller bearing is predicted to be 105 100 hours, compared with 'infinite' at 180 rev/min, the roller bearing is unlikely to fail during the life of this mechanism. As before, boundary lubrication occurs between the roller bearing and wheel slot faces throughout the entire indexing phase of the mechanism motion.

Consistent with the analysis for a crank speed of 180 rev/min, the 190 rev/min theoretical limiting speed of the mechanism is dictated by the maximum allowable contact stress. As expected, the contact stress rises above the allowable material limit when the mechanism operates at 250 rev/min. The software gives a warning that the predicted value of 1009 N/mm² exceeds the limit of 850 N/mm².

The warning (see Section 3.4.1) to check that the component has adequate strength in the 16 mm wide section between the slot roots and the hub is again considered. Increasing the speed from 180 to 250 rev/min raises the contact force at maximum torque by approximately 85% from 736.1 to 1367 N, applied to a point 77 mm radially out from the wheel shaft centre.

The finite element analysis described in Section 3.4.2 is repeated for the increased force and the maximum von Mises stress rises from 23.7 to 64.6 N/mm². As before, structural (*i.e.* fatigue) failure of the wheel is highly unlikely for the calculated stresses.

3.6 Reduction of Contact Stress

Since the allowable contact stress limit is exceeded if the motor operates at a speed above 190 rev/min, it is necessary to try to improve the design. The contact stress and bearing performance are dependent on the contact force, the bearing dimensions and the material properties of the slot face and bearing. See Data Item 96002 (Reference 2).

In this example, the bearing diameter cannot be changed since the slot width is fixed as 30 mm. Changing the material properties of the slot faces requires design modification of the wheel and may be expensive. It can be shown from Equations (3.54) to (3.58) of Data Item 96002 (Reference 2) that increasing the crown radius of the roller reduces the contact stress. If possible, however, the most cost-effective solution is to reduce the contact force.

The roller bearing performance would also benefit from a reduction in the contact force. Although the predicted life of 105 100 hours is more than twice the 50 000-hour minimum, the calculation is based on theoretical assumptions that may not be entirely reproduced in practice. It is therefore desirable that the margin between the predicted value and the stated limits is maximised within economic constraints.

Finally, in addition to improving the life and reliability of the roller bearing, a smaller contact force reduces the power requirement from the motor.

The efficiency of the design process can be maximised by using ESDUpac A9611 to identify the dominant factor (if any) that dictates the value of the contact force.

3.7 Identifying the Dominant Factor Causing the Maximum Contact Force

Contact force is related to the total torque loading driven by the wheel at a particular speed. This torque is the sum of the inertial torque due to the acceleration of the wheel and the external loading applied to the shaft, but does not include the torque required to accelerate the crank. See Section 6 of Data Item ME3 (Reference 1).

To assess the significance of the external torque, the data specification in analysis 9901EX1B is revised. The new analysis is named 9901EX1C and both the mass and mass moment of inertia of the wheel are set to negligible values (1.0E-10).

Output from the program for this analysis shows that the maximum contact force is reduced from 1409 to 142.3 N which implies that approximately 10% of the contact force is due to the external torque.

However, Section 6 of Data Item ME3 shows that the interdependency between the inertial effects, external load and contact force is complex and usually non-linear. A second check is therefore carried out by re-setting the external torque from "Load entered via file" to "No load" in the data specification for analysis 9901EX1B. The inertial values specified in Section 3.2 are used and the analysis named 9901EX1D.

Output from the program shows that the maximum contact force is reduced from 1409 to 1363 N, which implies that approximately 97% of the contact force is due to the inertia of the wheel and shaft.

It is therefore logical to assume (in this example) that the inertial torque dominates the contact force in the Geneva mechanism. The design of the wheel is therefore studied with the objective of reducing its mass and mass moment of inertia. Any removal or relocation of material must not significantly weaken the ability of the wheel to withstand the cyclic structural loads caused by the contact force.

While the results of analyses 9901EX1C and 9901EX1D imply that the external torque is a minor factor compared to the influence of the wheel inertia, its origins should always be reviewed. In the machine described in this example, the inertial properties of the components moved by the linkage dominate the external load. The sizes and masses of these components are fixed. Any changes made to the mechanism links to reduce their inertial properties will not significantly affect the external load acting on the Geneva mechanism - but will probably reduce their stiffness and strength.

3.8 Options for Increasing the Limiting Speed of the Mechanism

Reducing the inertial properties of the wheel or its shaft is not possible and the roller bearing diameter is fixed as 30 mm. The feasibility of increasing the crown radius of the roller or hardening the wheel slot face material are both considered and their contributions toward the development of a cost-effective engineering solution assessed.

3.8.1 Non-standard crown radii

Consideration of the contact stress equations presented in Section 3.4 of Data Item 96002 (Reference 2) shows that increasing the crown radius of the roller reduces the contact stress and increases the contact area between the roller and slot faces. It is desirable to maximise the contact 'footprint' while avoiding an overlap of the edges of either the slot face or the roller bearing. Since most commercially available roller bearings are supplied with a standard 500 mm crown radius (the default value specified in ESDUpac A9611) choosing a larger crown radius will increase the cost.

The 500 mm crown radius specified in analysis 9901EX1B is increased to 1000 mm and the new analysis is named 9901EX1E. Output from the program shows that the contact stress is reduced from 1009 to 875.8 N/mm² with a corresponding increase in the length of the contact ellipse from 2.52 to 3.32 mm. Output from the software gives no warnings, beyond those described in Section 3.4.1, and the design appears to be an acceptable solution to the problem. However, the cost and availability of a non-standard bearing size must be investigated (see Section 3.8.2) before this change is accepted.

Further benefits are predicted for even larger crown radii. For instance, increasing the crown radius to 40 metres (analysis 9901EX1F) increases the predicted length of the contact ellipse to 13.5 mm and reduces the contact stress to 433.3 N/mm². Unfortunately, although the contact zone theoretically remains within the width of the roller bearing and slot, the cost and difficulty of manufacturing a bearing with this radius within acceptable manufacturing tolerances makes this an unrealistic specification.

Further, the accuracy of the equations used to predict the contact stress and the size of the contact zone may be questionable unless the size of the contact zone is very small compared to the roller radius. See Section 3.4 of Data Item 96002 (Reference 2).

Finally, the height of the crown of a 40 metre radius on a 14 mm wide bearing is less than one micron. Misalignment in the mechanism and/or deformation of the crown under load would negate any benefits gained. The manufacturing tolerances on the dimensions of the contacting components would also be so tight as to be prohibitively expensive – if indeed they could be achieved.

Therefore, a compromise is made between the cost and difficulty of producing a non-standard bearing with a very large crown radius and the benefit of a large reduction in the contact stress. A 2500 mm crown radius is specified and the analysis named 9901EX1G. Comparing this specification with analysis 9901EX1B, the contact stress is reduced from 1009 to 731.1 N/mm² and the length of the contact ellipse increased from 2.52 to 4.75 mm. Since the minimum width of the bearing and wheel slot is 14 mm, the contact ellipse is not expected to overlap the edges even if misalignment exists between the wheel and roller bearing axes.

With this crown radius, the limiting speed of the mechanism increases from 190 to 275 rev/min. At 275 rev/min, output from the program shows that the modified life rating for the roller is less than the required 50 000 hours.

3.8.2 Cost and availability of roller bearings with non-standard crown radii

Various manufacturers and suppliers of track roller bearings were contacted to assess the likely cost and delivery times for a 30 mm diameter track-roller bearing with a 2500 mm crown radius. Some organisations state that it is not profitable to supply any bearings with non-standard crown radii, whereas others require an order quantity exceeding 200 bearings. Other manufacturers were prepared to produce non-standard bearings but at three or four times the cost of standard bearings.

Crucially, delivery times are quoted in months and so the use of a non-standard crown radius is rejected.

3.8.3 Increasing the wheel slot face hardness

Since it is not cost effective to reduce the maximum contact stress by changing the dimensions of components in the mechanism, the allowable contact stress in the wheel slot faces must be raised. The existing wheel is modified to allow the use of slot face inserts that can subsequently be replaced, if required. By specifying a suitable heat treatment of the slot face insert surfaces to increase their hardness, the maximum allowable contact stress is raised from 850 to 1200 N/mm².

ESDUpac A9611 is run with the modified limit, a standard crown radius of 500 mm and the file names are changed to 9901EX1H. Compared with the initial analysis, 9901EX1B, the limiting speed is raised from 190 to 275 rev/min. As was the case for the analysis 9901EX1G (see Section 3.8.1), the modified life rating for the roller is less than 50 000 hours if the crank speed is greater than or equal to, 275 rev/min.

3.9 Problem Solution

There appear to be two cost-effective solutions to this design problem, if the specification of crowned roller bearings with non-standard crown radii is rejected (see Section 3.8.2).

First, the speed of the machine could be limited to 190 rev/min so that the allowable contact stress limit is not exceeded. This restricts the productivity increase expected for this machine and is not acceptable.

It is therefore logical to adopt the second alternative and harden the wheel slot surfaces as discussed in Section 3.8.3, thereby raising the limiting speed of the mechanism from 190 to 275 rev/min. This is the recommended solution with data specification 9901EX1H.

Before accepting this design solution, however, the effect on the performance of the mechanism of the expected speed variation shown in Sketch 3.1 is assessed.