DYNAMIC POSITIONING CONFERENCE
September 18-19, 2001

THRUSTERS SESSION

Bevel Gears for Thrusters

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1. Introduction

For more than 70 years right angle gear boxes have been used for marine drives. One of the first was built by the company Voith in Germany, the Voith-Schneider® Propeller. After the Second World War the company Schottel came up with one of the first thrusters driven by two bevel gear sets. Nowadays installations are running for instance in Finish ice-breakers with a power of 7500 kW which are driven by right angle gear boxes.

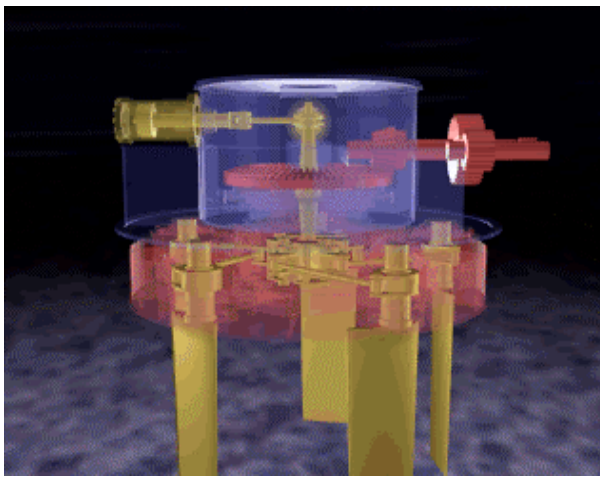

Figure 1: Voith-Schneider® Propeller

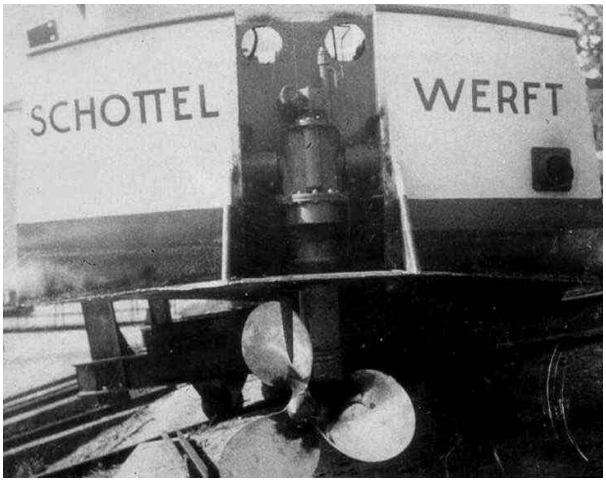

Figure 2: The first Schottel Thruster

For more than 30 years the Klingelnberg company has produced spiral bevel gears for right angle gear drives. Klingelnberg Söhne GmbH, a member of the Klingelnberg Oerlikon Group, is an international active machinetool manufacturer in Germany. The company employs 800 persons in its production basis in Hückeswagen near Cologne and in Ettlingen near Karlsruhe and has an annual turnover of approximately 170 million Deutsche Mark. The activities of the Klingelnberg-Oerlikon Group whose origin dates from 1814 is managed today by Dieter Klingelnberg who is the chairman of the board of directors. The Klingelnberg Söhne GmbH in Hückeswagen is engaged in spiral bevel gear cutting machines for small and medium sized production series, spiral bevel gear grinding machines, worm and thread grinding machines, hob sharpeners, gear measuring machines and job manufacturing of spiral bevel gears. In our spiral bevel gear job shop we produce spiral bevel gears in a range from 10mm up to 2200mm pitch diameter. Since 1980 we have been able to hardskive case hardened spiral bevel gears within the German DIN Quality of 6 or the AGMA Quality of 12. The spiral bevel gear job shop has an annual turnover of approximately 30 million DM. Nearly 60 – 70 % of all spiral bevel gears which we have produced in the last years are for right angle gear boxes in marine drives. The material of all these gear sets is a high-grade forged steel. The complete production process is attended by classification societies like Det Norske Veritas, ABS, Bureau Veritas etc.

2. Examples for Right Angle Gear in Thrusters

There are different possibilities to drive the propeller of a ship. The most typical one is with a vertical motor connected with a coupling to a vertical pinion shaft, which drives a horizontal propeller shaft. This system is used for nearly all tunnel thrusters.

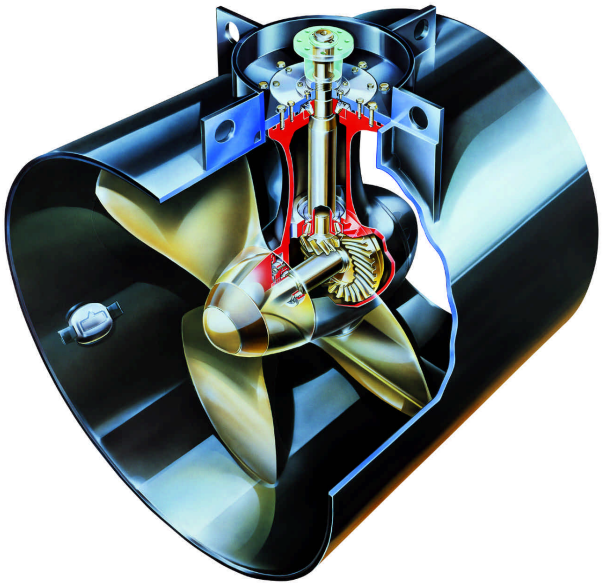

Figure 3: The principle of the tunnel thruster

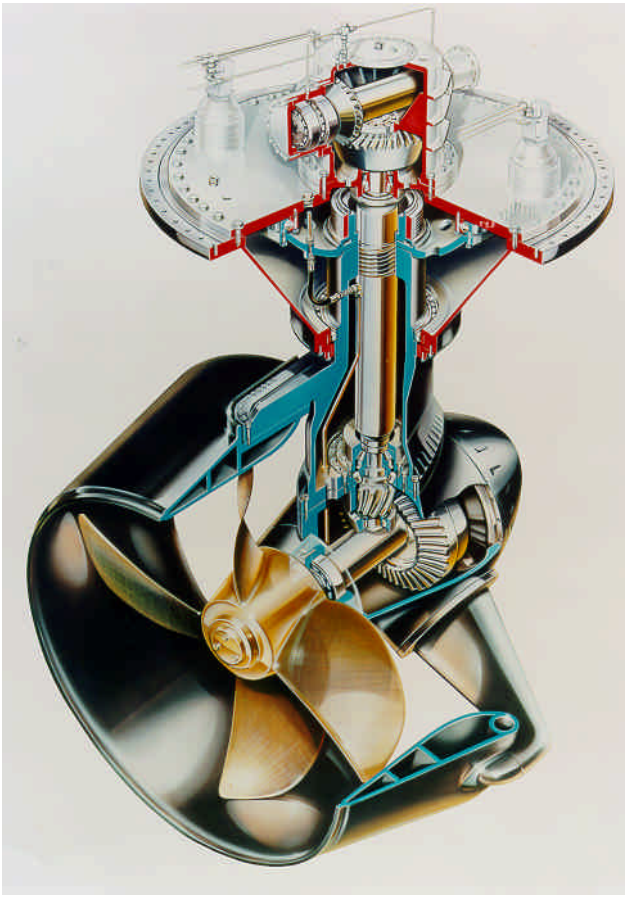

Figure 4: The principle of a Z-drive

Another possibility is the mostly used one for dynamic positioning, the so-called Z-drive (figure 4). In this case, a horizontal electric motor or diesel engine drives a top bevel gear set which transmits the power to an underwater spiral bevel gear set which is connected directly to the propeller. The whole unit can be moved 360° around its vertical axis. That allows a very high grade of maneuvrability.

But there are also other possibilities known which can be seen in figure 5 .

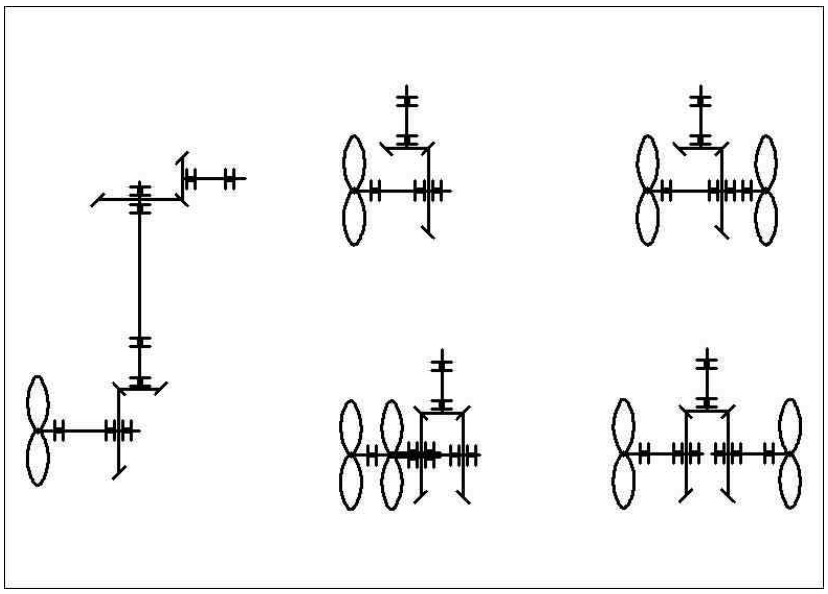

Figure 5: Examples for right gear in thrusters

For instance one pinion is in mesh with two gears facing each other. That is called the Contra Rotating System (figure 6). Or a spiral bevel gear set is driving one output shaft with two propellers, which is called *Twin Propeller*.

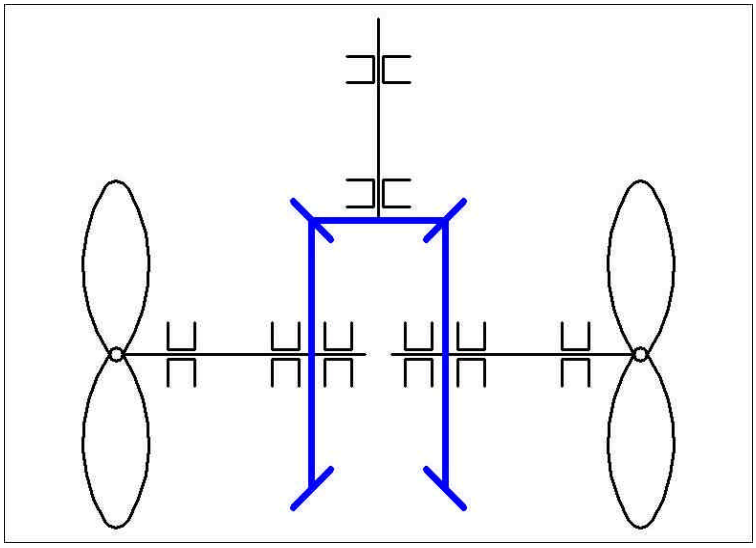

Figure 6: The Contra Rotating System

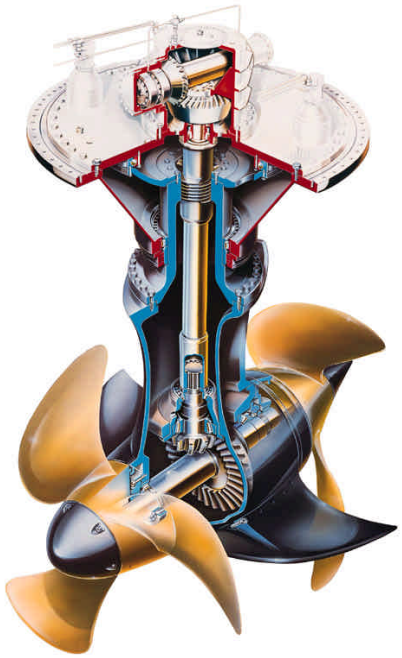

Figure 7: Twin Propeller

For all these gear drives it is very important to produce high precision spiral bevel gears, which are able to compensate all deflections which may occur under full load. There are a couple of influence factors like thermal expansion and elastic deformation of the housing or propeller shaft or even the bevel gear teeth.

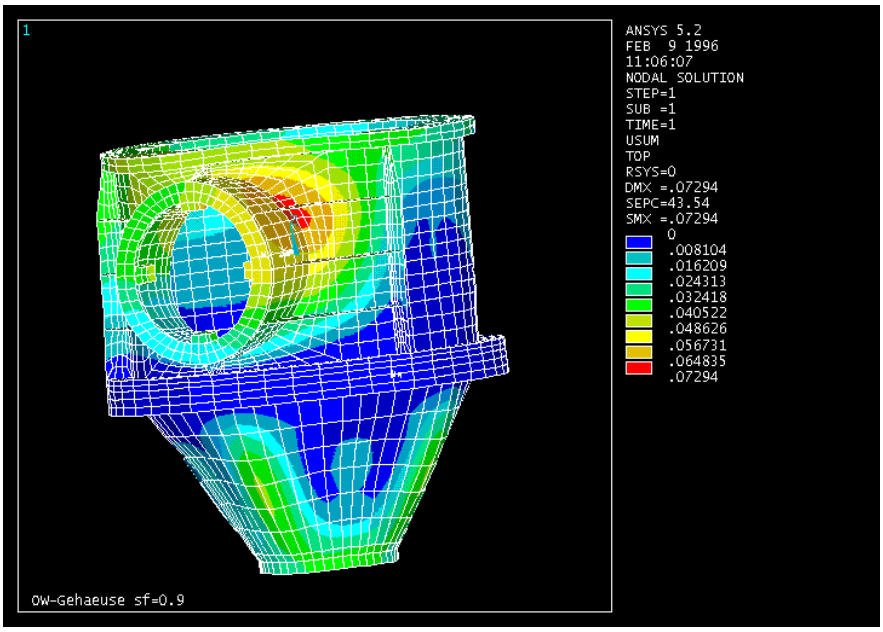

Figure 8: Calculated deformation of a housing

3. Characteristics of Spiral Bevel Gears

In the following the basic characteristics of spiral bevel gears are explained. In the figure to the right you can see the typical shape of a spiral bevel gear tooth produced according to the Klingelnberg Cyclo-Palloid method. Typical is the constant tooth depth along the face width and an approximately constant normal tooth thickness. Pinion and wheel are produced in a continuous cutting process. I. e. the tooth gaps are not cut one after the other but cutter head and workpiece rotate simultaneously while the blades run through consecutive gaps. This method is called face hobbing. Therefore, the lengthwise curvature is an epicycloid.

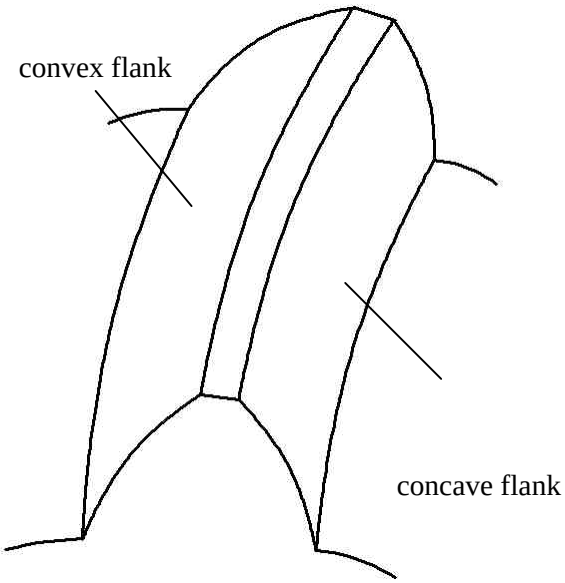

Figure 9

As mentioned already, in a bevel gear box there is much deflection under load. So, gear teeth, which would have line contact along the face width when running together without load will end up with hard contact at the toothend as soon as the gears are loaded. Therefore, it is necessary to give meshing bevel gears flanks a crowning also called mismatch, which results in a limited contact pattern within the face width. That means for spiral bevel gears, to produce a little less curvature on the concave flanks than on mating convex flanks. With rotating cutter heads this is achieved by different operating radii of the blades cutting the concave flanks and the convex flanks (figure 10).

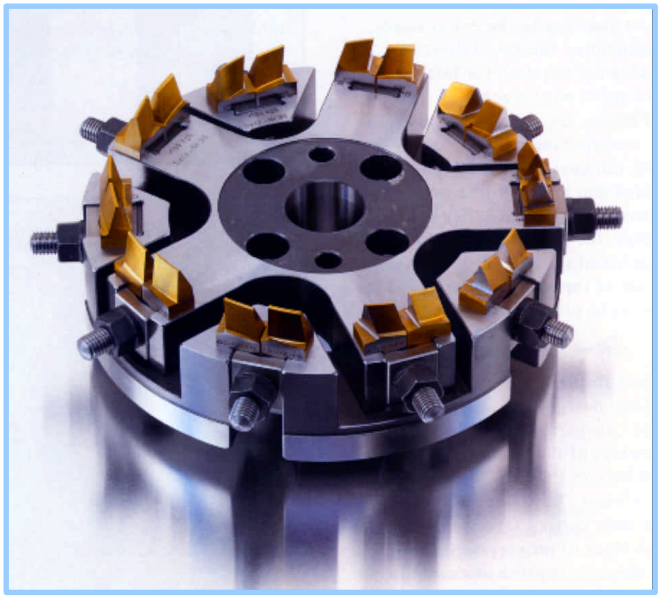

Figure 10: Two part cutter head

There are different systems for spiral bevel gears how to generate this lenthwise crowning by different radii. We think the most flexible and easiest way to handle is the Klingelnberg Cyclo- Palloid system. In figure 10 you can see a typical “Two- part” cutter head. On one part there are all inner blades which cut the convex flanks. In order to achieve this in a continious cutting method the two parts of the cutter head rotate around parallel axis which are set to a small distance on the cutting machine. This amount of “excentricity” EX is given by the difference of radii for the inner blades and outer blades as shown in figure 11 so that the length of the contact pattern can be adapted to the relevant operating conditions. The two- part design of the cutter head also permits independent corrections of the contact pattern on both flanks and it is easily possible to shift the pattern closer to the toe or the heel if required.

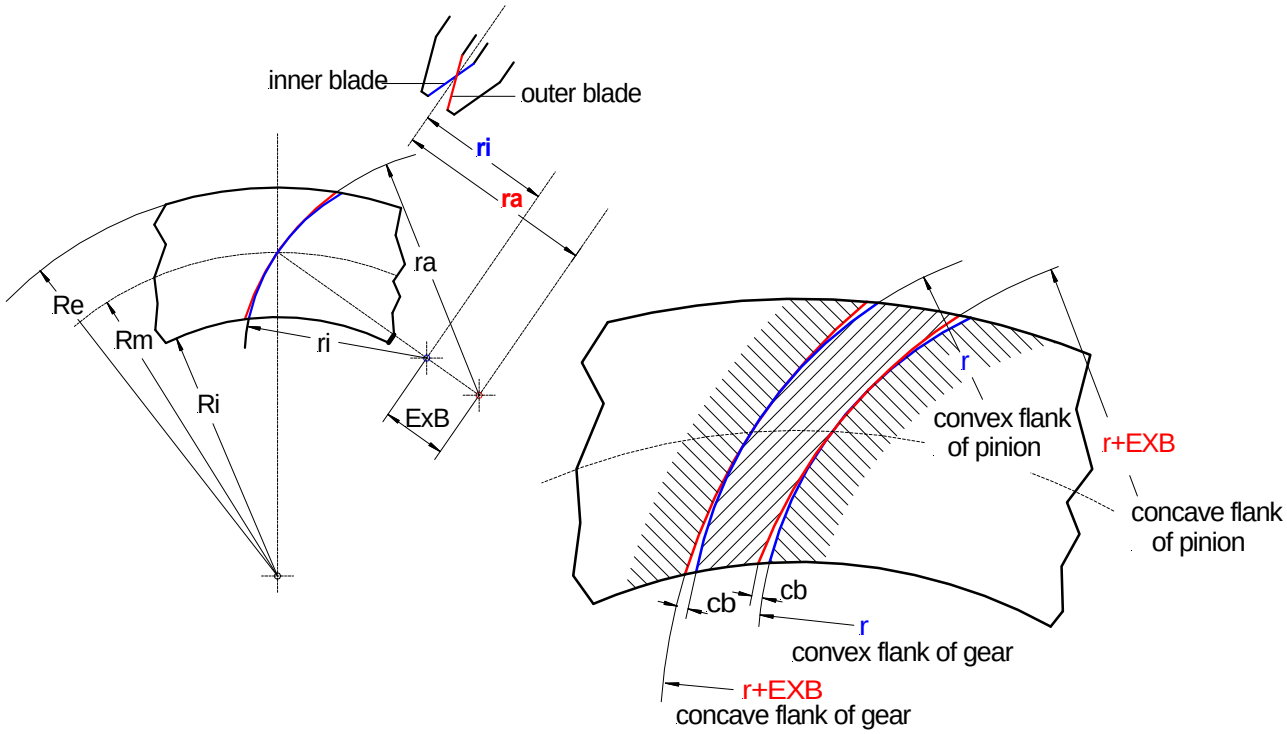

Figure 11: Generation of the lenthwise crowning

The two-part design of the cutter head permits to correct the length of the tooth bearing pattern. This is possible because all blades for the convex flanks are arranged on the inner cutter head and all blades for the concave flanks on the outer cutter head. Both parts of the cutter head rotate about axes which are spaced apart by the excentricity distance EX. The effective cutter radius r of the outer cutter head is thus increased by the amount EX, whilst radius r of the inner cutter head remains nominal. Thus, the concave flanks of pinion and gear are less curved than the convex flanks which produces a tooth bearing pattern of a limited length when convex and concave flanks mate. Because EX can be selected quite freely within customary limits, one can adapt the length of the tooth bearing pattern to relevant operating conditions. The two-part design of the cutter head further permits independent corrections of the tooth bearing pattern on both flanks.

Like the lengthwise crowning it is also necessary to create a crowning vertical to the tooth length in order to prevent that the tip of the pinion tooth or the tip of the gear tooth gets into hard contact due to elastic deformation of the gear teeth. This profile crowning is created by a radius, which is ground into the profiling side of the cutter blade. Figure 12 shows the hollow shape in such a blade.

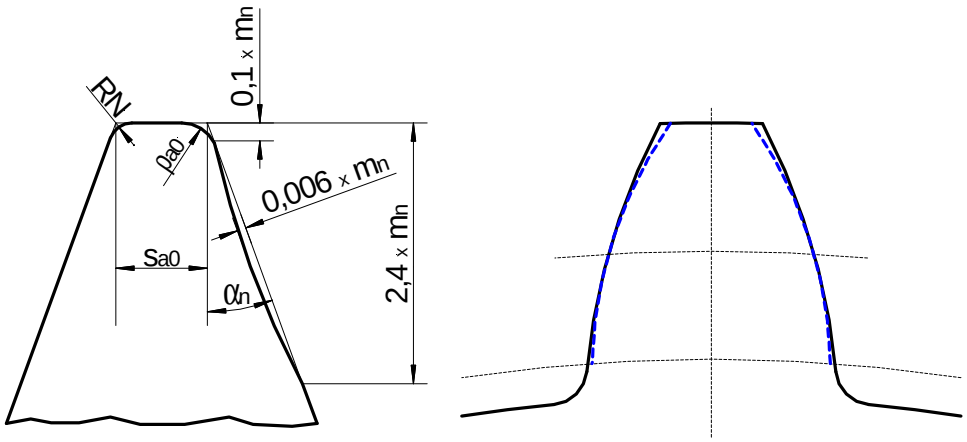

Figure 12: Generation of profile crowning

The standard amount for the depth of the hollow shape is 0.006 times the blade module. This special profile in the blade allows that more material from the tip and from the root of the tooth is cut off while the theoretical thickness of the tooth is constant. Both effects, the lengthwise and the profile crowning, end up in a theoretically elliptical contact pattern, which is localized in the centre of the tooth under normal circumstances. This contact pattern should have a size of 40 – 50 % of the tooth length and should spread under load up to 85 % of the face width. With that devided cutter head system it is easily possible to shift the contact pattern closer to the toe or to the heel if required.

The majority of all bevel gears we cut should have their contact pattern in the centre of the tooth. To get more detailed information a full load test can be taken which shows the amount of displacement between the loaded and unloaded contact pattern. This information can be useful for a correction that should end up with a load contact pattern, which is located in the centre of the tooth. Today, it is possible to calculate the deflections and thermal expansions of the housing under load by Finite Element Methods (FEM). The amount of these deformations can be expressed to x, y, z displacements of the pinion relative to the gear transformed to a calculation program like BECAL. There it is possible to show what happens with the contact pattern under load, which will be explained later.

4. Geometry Calculation and Rating

Every gear production starts with the geometry calculation and the rating. In figure 13 a typical gear dimension sheet of the Klingelnberg program KN 3029 is shown. There is a lot of experiences from more than 30 years available to create a good gear design depending on the application of these gears. It always starts with the dimension of the gear set, which should not exceed a given housing or with the input torque or input horsepower. A maximum possible face width and a maximum profile and spiral overlap guarantee a low noise level and a maximum load capacity. The rating of Klingelnberg spiral bevel gears is done according to our rating program called KN3030, which is based on DIN 3991. In former times these ratings were done according to the Lewis formula and the Hertzian stress. Roughly 15 years ago we started with the DIN calculation. Meanwhile there are a lot of experiences with that calculation system which makes sure that we are producing bevel gears on the safety side.

```
     K L I N G E L N B E R G   C Y K L O - P A L L O I D
               G E A R I N G   G E O M E T R Y
PROGRAM  K3029                                              PAGE 1
C-NO      4711 -Example                        -   DATE 13. 8.2001
- - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - -
I N P U T   D A T A  (METRIC SYSTEM)

SHAFT ANGLE                  SIGMA   =           90.00

OFFSET                       AV      =           +0.00

NUMBER OF TEETH              Z       =   13              42
NORMAL MODULE     MEAN       MN      =           14.8176 CALCULATED

PITCH DIAMETER OF GEAR       D0 2    =          925.00
FACE WIDTH                   B 2     =          173.00
SPIRAL ANGLE                 BETA M  =   35.000             35.000

CUTTER RADIUS                R       =          268.00
CUTTER BLADE MODULE          M0      =           14.00
NUMBER OF STARTS CUTTERH.    Z0      =    5               5

PRESSURE ANGLE  DRIVE/COAST  ALPHA   =   20.00             20.00

ADDENDUM MODIF. FACTOR       X1  FOR BALANCED SPECIFIC SLIDING
TOOTH THICKN. MODIF. FACTOR  XS      =           0.014 CALCULATED
ANGLE MODIFICATION           TETA K  =            0.00

STUB FACTOR                  HAP *   =            1.00
- - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - -
RATIO OF NUMBER OF TEETH     U       =           3.231

PITCH CONE ANGLE             DELTA 0 =   17.199             72.801


TRANSVERSE MODULE OUTER      MTE     =   22.02              22.02
NORMAL MODULE     OUTER      MNE     =           14.29
                  INNER      MNI     =           13.34

OFFSET ANGLE                 ZETA M  =            0.00
LIMITING PRESSURE ANGLE      ALPHA G =            0.00
- - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - -
V I R T U A L   C R O W N   W H E E L   D A T A
NUMBER OF TEETH              ZP      =         43.9659
BASE CIRCLE RADIUS           RHO     =          327.46
CONE DISTANCE     OUTER      RE      =  484.15          484.15
                  MEAN       RM      =  397.65          397.65
                  INNER      RI      =  311.15          311.15

SPIRAL ANGLE      OUTER      BETA E  =   49.54              49.54
                  INNER      BETA I  =   19.49              19.49

CROWN WHEEL:         OFFSET  AV'     =            0.00
               OFFSET ANGLE  ZETA'M  =            0.00

MACHINE DISTANCE             MD      =          364.70
- - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - -
  MD     FK 41B  KNC 25  KNC 40/60  AMK 855  AMK 1602  AMK 1604/5
MIN/MAX    0/70   0/150      0/290    0/460   250/900    350/1000
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     K L I N G E L N B E R G   C Y K L O - P A L L O I D
               G E A R I N G   G E O M E T R Y
PROGRAM  K3029                                              PAGE 2
C-NO      4711 -Example                        -   DATE 13. 8.2001
- - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - -
C H E C K I N G   V I R T .  C R O W N   W H E E L

SLOT WIDTH AT ROOT (NORMAL)  WITHOUT BACKLASH
  AT   RY2 = 421.50          EFN MAX =    9.49              10.34
       RE2 = 484.15          EFNE    =    8.54               9.39
       RI2 = 311.15          EFNI    =    7.05               7.90

TOPLAND OF UNIVERSAL BLADE   SA0     =              4.82
CUTTER EDGE RADIUS           RHOA0   =    4.20               4.20
                             RHOA MAX=              5.07

ADDENDUM MODIF. FACTOR       X1      =              0.37

  AVOIDING UNDERCUT          XI MIN  =              0.26


- - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - -
MEAN PITCH CONE DIAMETER     DM      =  235.16          759.74
VIRTUAL NUMBER OF TEETH      ZN      =   23.42          244.40

TRANSVERSE CONTACT RATIO     EPS A   =              1.17
FACE CONTACT RATIO           EPS B   =              2.14

TOOTH FORM FACTOR  DR./CST.  YFA     = 2.23/2.23        2.23/2.23
STRESS CORRECTION FACTOR     YSA     = 1.85/1.85        1.91/1.91
 ACCORDING TO DIN 3991
- - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - - -
G E A R   D I M E N S I O N S

WHOLE DEPTH                  H       =             33.34
ADDENDUM          MEAN       HA      =   20.30              9.34

PITCH DIAMETER               D0      =  286.31          925.00
FACEWIDTH                    B       =  173.00          173.00


                             DAE     =  325.09          930.52
                             DAI     =  222.79          599.99
                             DAIK    =  222.79

                             LH      =  462.50          143.15

                             LA      =  165.26           51.15
                             LAK     =  165.26
                             LW      =  291.23           83.08
                             LWK     =  291.23
                             LD      =  379.87          117.58

TIP RADIUS AT TOOTH END      RR      =              5.0


A X I A L   L O A D S  FA  PERCENTAGE OF FMT (WITHOUT SLIDING)
CONCAVE PINION FLANK CONTACTS     FA1 =  80.0   FA2 =  21.7   DRIVE
CONVEXE PINION FLANK CONTACTS     FA1 = -53.8   FA2 =  63.1   COAST
```

Figure 13: Geometry calculation sheet

In the following pages we give some brief information about the basic background of the rating according to the Klingelnberg program KN 3030 placed on the standard according to DIN 3991.

- **Calculation formulae**

  In order to calculate the load capacity of spiral bevel gears, the following method produces usable results:

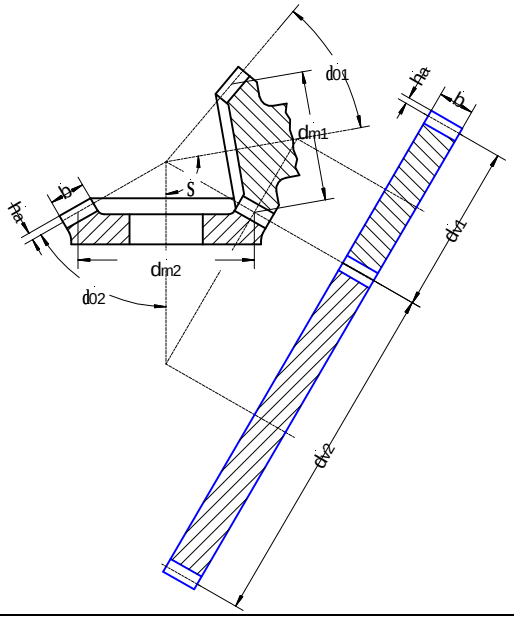
  In its meshing area, the tooth rim of a bevel gear is assumed to be that of a cylindrical gear, and the spiral teeth are regarded as helical teeth. The involute profile on the cylindrical gear is very close to the bevel gear tooth.

  For the virtual cylindrical gears which are found, it is then possible to use the well-known rules of standard DIN 3990 to calculate contact stress, bending stress and scuffing temperatures. By means of an additional correction factor, use of DIN 3991 is made to assume that the same stresses accor with the spiral bevel gears as with the virtual cylindrical gears and that the same allowable values apply.

  When designing a bevel gear stage, it is necessary to determine the main criteria such as power rating, gear ratio, machine size, etc., as well as a number of factors which influence the results of calculations in various ways and to differing extends.

- **Geometry of virtual teeth (Tredgoldsche Näherung)**

  At mid-face of the bevel gear tooth a transverse section is positioned, into which a virtual cylindrical gear is fitted. The axis of this cylindrical gear runs parallel to the common reference cone line and passes through the point at which the transverse section intersects the bevel gear axis. Figure 14 shows the relationship between the dimensions of the bevel gear and the dimensions of the virtual cylindrical gear.



- **Load values**

The nominal torque of the operating machine is decisive. This is the operating torque to be transmitted over a longer period of time and under the most severe regular working conditions. As a substitute, it is also allowed to use the nominal torque of the driving engine if it corresponds to the required torque of the operating machine.

- **Basic equations for rating**

A comparison of two stress values is performed to demonstrate that a design is sufficient against failure. Using the actual forces and the geometry of the design and on analogy with the beam theory resp. with Hertzian contact stresses, a so-called local stress value is calculated. Then a failure stress value is to be determined on test gears. This failure stress value is set into relationship with the local stress value and adapted to the performance conditions.

As long as this relationship is not below the valid safety value, then the durability is deemed to have been proved. The proof must be performed for the flank in respect of pitting durability and for the tooth root in respect of breakage, where necessary separately for pinion and gear. Similarly, in the proof for scuffing resistance, a ratio of actual and allowable temperature is formed.

Pitting durability

$$S_H = \frac{s_{HG}}{s_H} > S_{Hmin}$$

Bending strength

$$S_F = \frac{s_{FG}\, Y_A}{s_F} > S_{Fmin}$$

Scuffing resistance

$$S_S = \frac{\Theta_{Sint}}{\Theta_{int}} > S_{Smin}$$

$S_{H(F,S)}$ = Calculated Safety

$s_{H(F)G}$ = Limit Load

$s_{H(F)}$ = Actual Load

$S_{H(F,S)\,Mim}$ = Minimum Safety Factors

Θ_{Sint} = Allowable Scuffing Temperature

Θ_{int} = Actual Integral Temperature

- **Safety factors**

The safety factors are affected by the material used, the method of manufacture and by the application. The following aspects should be taken into account:

Damage probability of durability values
Quality of workpieces
Quality of heat treatment
Tolerances in the manufacturing process
Tolerances during assembly procedure
Actual operating forces
Type of lubrication
Maintenance conditions

The objective of rating calculations is to ensure a high level of operating safety at a reasonable cost. The more carefully calculations are performed (i.e. the indicated influences are taken into account), the lower can be the safety value.

In this works standard, the following minimum safety factors are assumed in respect of durability:

Pitting	S_{Hmin} = 1.1 ... 1.2
Tooth breakage	S_{Fmin} = 1.5 ... 1.6
Scuffing	S_{Smin} = 1.8 ... 2.0

The responsibility for setting or evaluating the necessary safety values lies with the gear manufacturer or the user, who has to provide the bevel gear manufacturer with the appropriate information in respect of operating conditions. this means that higher safety values must be used if a higher risk of damage is to be expected.

There are of course a number of comparable rating calculation methods, like AGMA, and classification society programs basing on the same approximation of Tredgold and using their own influence factors. Also different safety factors are considered. There are some examples in the table below.

Table1: Standard safety faktors

Safety factor against	ABS	DNV	KS
Pitting	1,15 ... 1,4	1,15 ... 1,2	1,1 ... 1,2
Tooth breakage	1,4 ... 1,8	1,4 ... 1,55	1,5 ... 1,6
Scuffing	-	1,3 ... 1,5	1,8 ... 2,0

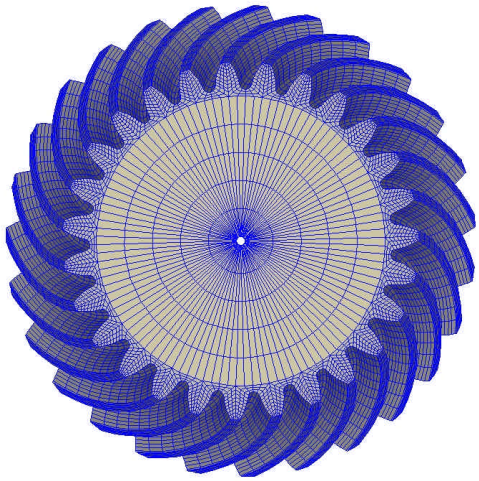
5. BECAL a new calculation program for Spiral Bevel Gears

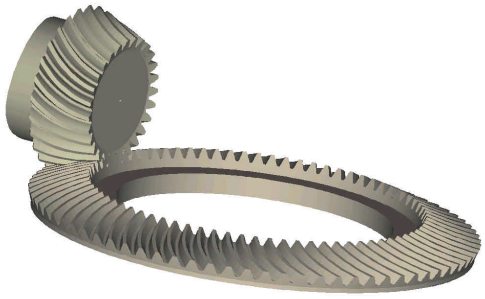
[2]A New Methodology for the Calculation of the Geometry, the Contact Pattern and the Gear Load Capacity of Bevel Gears

The standardized calculation methods DIN 3991 and AGMA 2003 make compromises in the proof of load capacity on the basis that the spatially complex geometry of bevel gearings with a curved flank line and their meshing conditions can only be (roughly) approximated by the Tredgold approximation model of the calculation method.

Essential properties that are specific to bevel gears, such as the variable geometry in the face width direction or the sensitivity to displacement, do not enter the calculation. A tooth bearing of known position and size is assumed. Compared with DIN 3991, the new draft ISO DIS 10300 is an improvement, since for the determination of the transverse and face contact ratios here an elliptical tooth bearing formed by profile bearing and crowning is applied which better comes up to the actual conditions. Precise calculation methods require a detailed knowledge of the flank geometry, which, however, is dependent on the manufacturing technique.″ [2]

″At the Institute of Machine Elements and Machine Design (IMM, Prof. Heinz Linke) of Dresden University of Technology such a device, the bevel gear calculation program BECAL, has been developed in cooperation with the Institute of Geometry (IfG, Prof. Gert Bär). In charge of the development is the Bevel Gears working group of the *Forschungsvereinigung Antriebstechnik e. V*. (research association for transmission technology, representative: Dr. H. Müller, Klingelnberg Söhne GmbH). ″ [7]





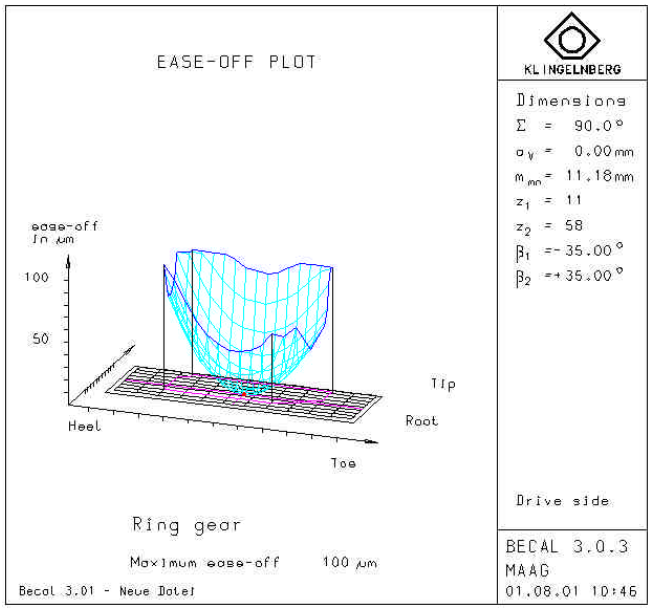
With this BECAL programme the following functions can be solved.

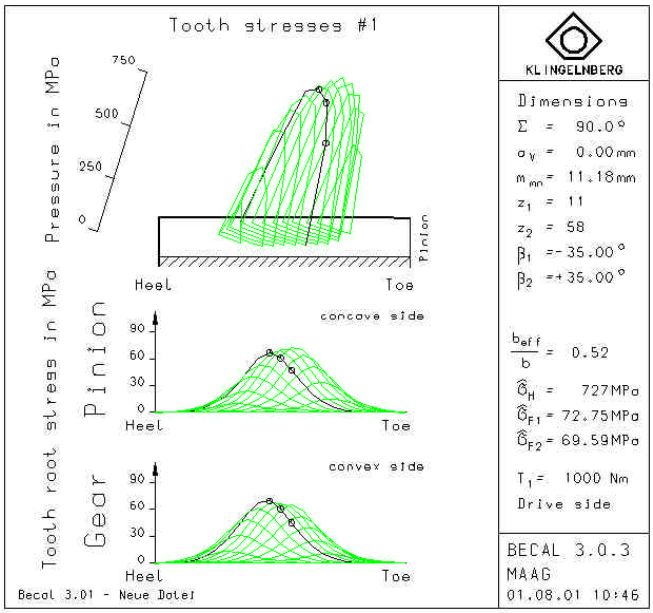
1) Exact tooth geometry

Calculation of the exact tooth flank and tooth root, geometry and topography using the original machine setting data of the gear producer.

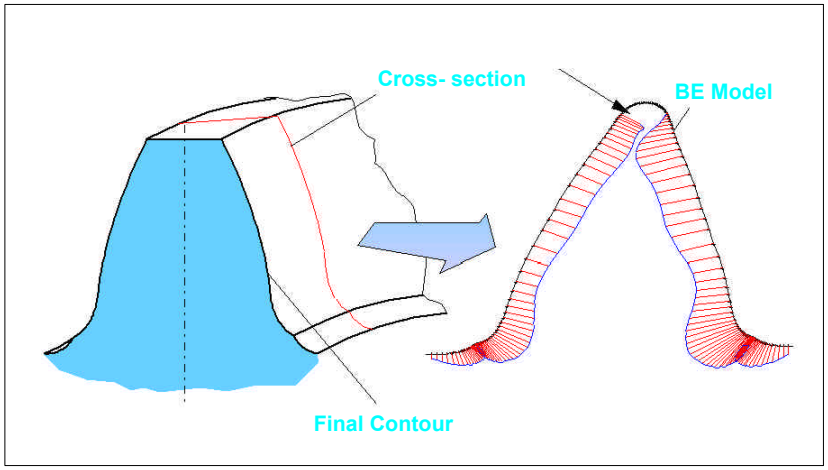
2) No load contact analysis

Beside the geometry of gearing pitch errors as well as the relative position of pinion and gear are included.





3) Calculation of the exact notch geometry in the tooth root



4) The influence of relative displacements between pinion an gear

5) Calculation of load distribution

The tooth root is subdivided into discrete segments in the face with direction. Their meshing conditions are the distances between the opposite flank points are calculated.

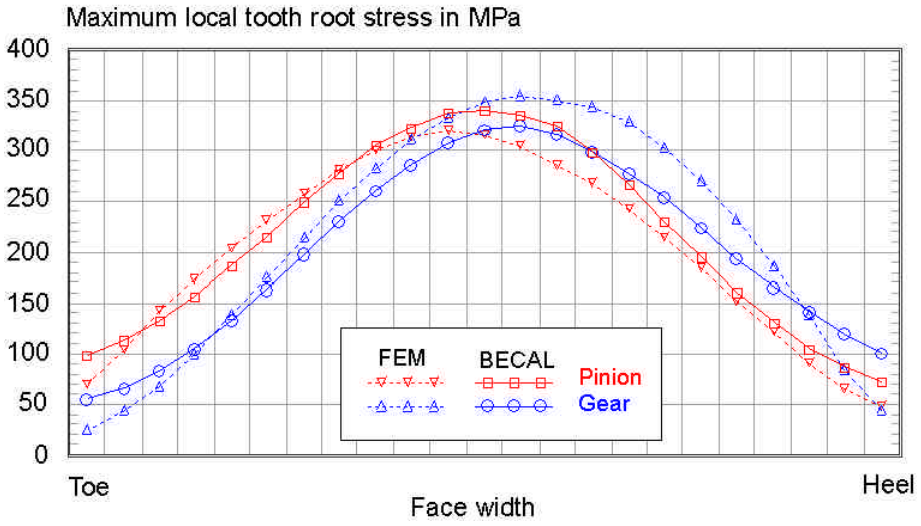
6) Flank contact pressure

The contact stress is determined of the tooth segments using the known formulae after Herz.

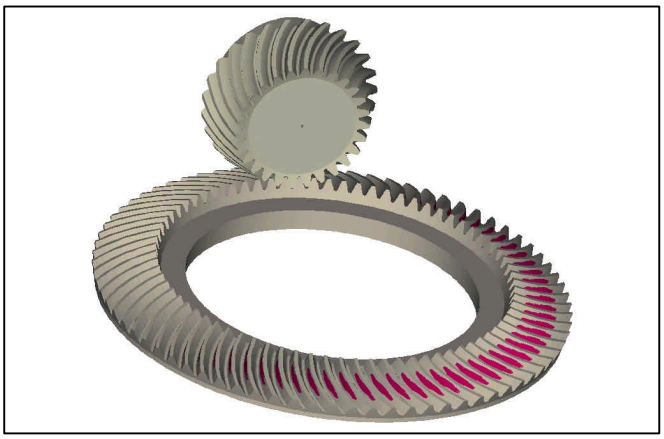
7) The local tooth root stress

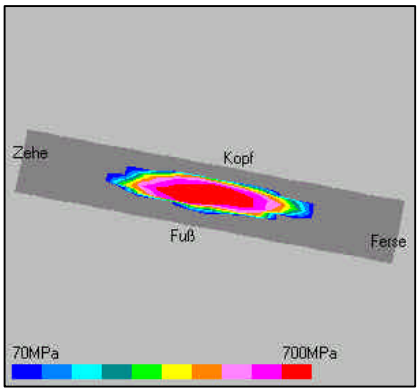
For precise determination of stress the Boundry Element Method (BEM) was chosen.

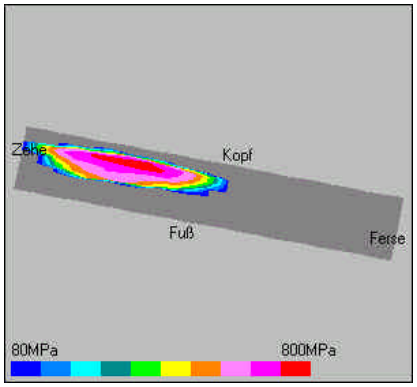
″The differences between the tooth root stresses calculated with the FEM (Finite Element Method) and with stress influence coefficients (BEM), are relatively small (about10 %) and lie within the range to be expected. A comparative representation of the results of the calculations is shown in Fig. for an example. ″ [7]



8) Load contact pattern







″BECAL can be used for different fields of applications:

- Specification of the basic geometry in the phase of design, i.e., the designer is assisted starting from the torque to be transmitted, the transmission ratio and the space taken up.
- Design of the environment with respect to its effects on meshing conditions and stress, and thus specification of changes of the environment (e.g. of the bearing).
- Specification of crowning
- Use for calculation on already existing gearings
- Assessment of cases of damage and the purposive determination of possible corrections
- Engineer-like assessment of stress

BECAL is used by German and international enterprises of different branches of industry.

At present, the design and optimization of spiral bevel gearings is still determined very much by the individual engineer´s experience.

With the calculation program BECAL there is a possibility of optimization of spiral bevel gears.

Objectives of optimization are the maximum safety of the gearing or highest stressability, insensitivity to displacements as well as low noise excitation.″

6. Outlook

In the future right angle gear boxes for marine drives and in general should be no longer a Black Box due to the possibility of

1. Finite Element calculation of load-dependent deformation of the environment.
2. Exact topography meassurement of the bevel gears.
3. Rating by ISO standard
4. Load contact and stress determination with calculation program like BECAL.
5. Simulation of full load test during gear assembling.

References

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[2] TU Dresden — *A New Methodology for the Calculation of the Geometry, the Contact Pattern And the Gear Load Capacity of Bevel Gears*;
Internet www page at URL http://mw.tu-dresden.de/becal/paris.html

[3] Klingelnberg — Tools for the Cyclo- Palloid Method related to their respective fields of application

[4] Voith — Power Point Presentation CD

[5] Klingelnberg — Workstandards KN 3030

[6] Klingelnberg — Internet www page at URL: http://klingelnberg.com/en/fset_.html

[7] TU Dresden — *The Development of the Program BECAL- an Efficient Tool for Calculating the Stress of Spiral Bevel Gears*
Internet www page at URL http://mw.tu-dresden.de/becal/becal-china.html