PET WASTE PROCESSING DRILL

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Abstract: The paper intends to design a waste processing drill with NX CAD in view of facilitating the operation of pressing and forming PET packs. It is made of a housing with two shafts inside having drilling and drive disks, between which the PETs will go. The two shafts are kinematically linked by two toothed wheels ($z = 42$, $m = 5$), mounted outside the housing, and the shafts are driven by a kinematic chain made up of a 3 kW electrical motor, cylindrical redactor and short-link roller chain transmission.

Key words: drill, waste processing, design and functioning.

1. INTRODUCTION

The world is heading toward an era of conservation, recycling playing a significant part. This is not a new method; it came along with mankind, and became a certainty alongside with the industrial revolution. Redundant metals, ferrous and non-ferrous, became standard materials for metallurgy.

Waste paper can be brought back to its original shape of pulp, and transformed into new paper.

Today, this waste recycling industry is completely internationalized. It collects, classifies, processes and markets for a wide range of materials – secondary products from industry and waste materials.

These secondary materials are produced at certain minimum specifications and then sold to industry.

International trade with secondary materials is necessary to supply steelworks, foundries, paper works, textile industry, plastic materials and rubber industry, etc. with raw material for a more ecological production.

Without recycling, the circuit of materials in nature would cause a series of events without a logical resolution. The possible useful materials would become available and would not be kept as a possible resource.

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The code for polyethylene terephthalate bottles or flasks is PET. PET recycling is necessary since their natural decomposing in the surrounding environment takes 500 years due to their composition.

2. PET DRILL DESIGN AND FUNCTIONING

PET drills are mounted between the feeding funnel and the tank of a patching press, to facilitate the operation of pressing and forming PET ballot, as in Fig.1.

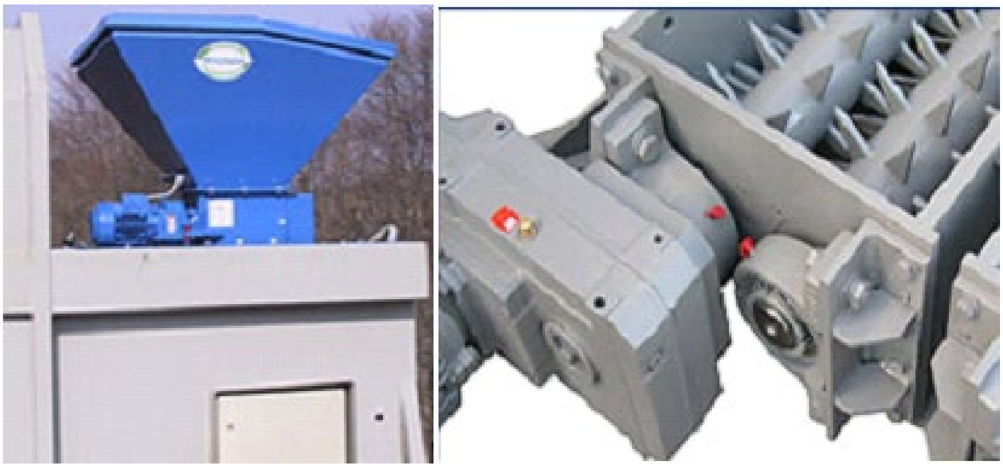

Fig. 1. Position of PT packs

With the help of NX 7.5 soft, a 3D PET drill has been designed and modelled, shown in Fig. 2.

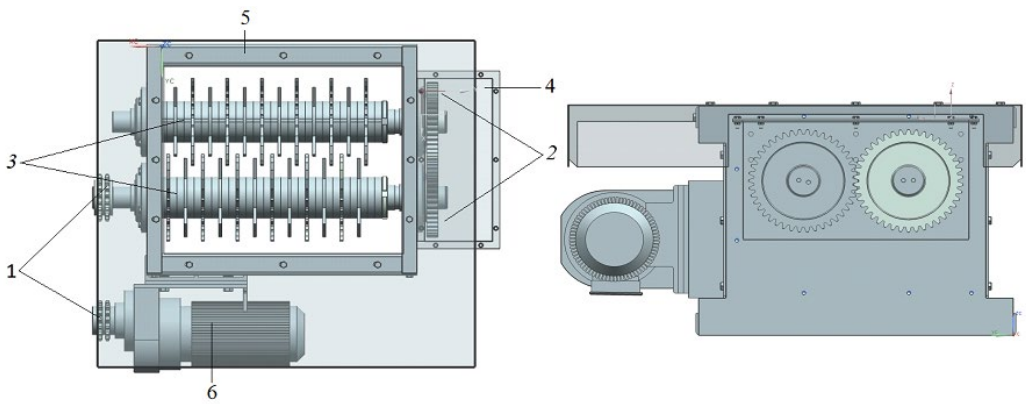

Fig. 2. PET drill

It is made up of a housing 5, with two shafts with drilling and driver disks inside 3, between which the PETS would pass. The two shafts are kinematically linked by two toothed wheels 2 ($z = 42$, $m = 5$), mounted outside the housing and protected by the oil reservoirs 4. The driving shaft is driven by a double roller chain 1 from a drive group 6. RFX77 – SEW drive group is made up of a 3 kW electrical motor and a cylindrical reduction gear of 10,88 transmission rate, achieving at the outlet shaft a 129 rpm rotation, 225 Nm momentum and being able to take over a 9030 N radial force.

The drilling block is made up of two shafts, one 883 mm long which is driven, and the other, 996 mm long which drives. Each have six driving disks and six drilling disks, separated by two distance rings, 26 in all, and fixed at the ends with two castellated nuts. The driving and drilling disks are fixed on the axis by the square shape of the respective bore hole of the axis and their constructive shape is shown in Fig. 3.

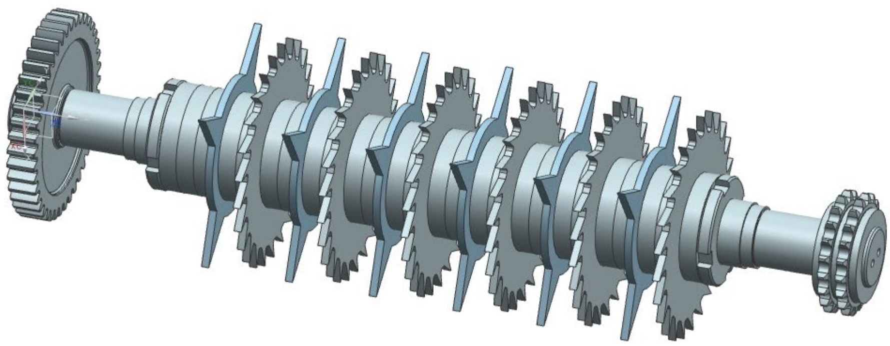

Fig. 3. Drive shaft

The ‘shredding’ block shafts rest on sliding bearings, which are fixed to the housing by four M16x40 screws. At the end of the drive shaft the double chain wheel is mounted, having z = 20 teeth, for roller chain and short-link double 12B, and at the other end the toothed wheel having z=42 teeth is mounted, as well as module m= 5 mm, both being fixed to the shaft by 12x8 mm parallel plane wedge each, and a blocking disk.

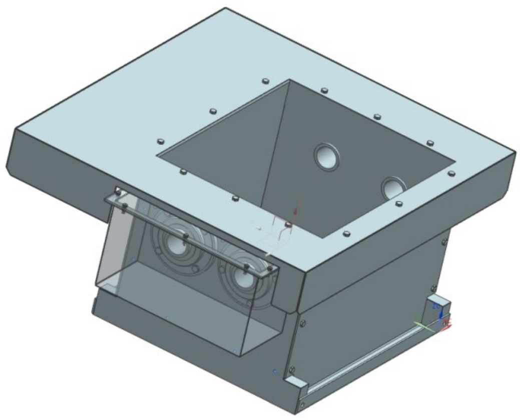

Fig. 4. Drill housing

The PET drill housing, shown in Fig. 4 is made up of two front plates, on which the borings for the gliding bearings are found, two lateral plates and the drill cover, which also serves as protection for the motor and chain transmission, these being fixed between them with the help of M12x30 screws.

The oil tank serves to protect and grease the toothed wheel gear, being hermetically locked by means of a gasket and a cover fixed by screws.

3. DIMENSIONING AND CHECKING THE KINEMATIC CHAIN OF THE DRILL

The kinematic chain of the drill is made up of a short-link roller chain 12B double, transmission rate being $z_1/z_2 = 17/20$ and a transmission by cylindrical toothed wheels with $z_3/z_4 = 42/42$ transmission ratio for an m = 5 mm module.

Transmission by links and rollers has a series of advantages: the shaft and the bearings take over only the reactions due to the momentums transmitted, since a stretching force at mounting is not necessary; constant transmission ratio; distances between relatively large axes and with no high precision. The disadvantages lie in the lack of evenness in the transmitted motion and the high noise produced by the impact between the teeth and the rollers.

Based on the literature MathCAD program has been devised allowing establishing the chain type used, the geometric elements of the two chain wheels and the transmission, the selection of manual greasing, determination of forces and tear, stretch and contact pressure check.

The safety coefficient for static tear should be higher or equal to 7, in the case of PET drills, this condition is checked by the equation:

$$\gamma_{st} = \frac{S_r}{F_a} = 13{,}901 \tag{1}$$

where: S_r is the the chain tear force, S_r = 59000 N; F_a – force on the drive branch, F_a = 4244 N.

The γ_D safety coefficient for dynamic tear should be higher than or equal to R and is determined with the equations:

$$\gamma_D = \frac{y\,S_r}{F_t} = 8{,}758 \quad R = \gamma\,\lambda_D = 10{,}366 \quad \frac{\gamma_D}{R}\,100 = 84{,}486 \tag{2}$$

where: y is the shock coefficient, y = 0,63; F_t – tangent force, F_t = 4244 N; γ – tear safety coefficient, γ = 12,55; λ_D – correction factor, λ_D = 0,826.

It is noticed that this condition is not met, which leads to reduction of functioning time from 10000 hours to 3000 hours.

Stretching verification of the chain is met, and its value is smaller than the value of the force allowed by the reducer 9030 N.

$$C_v = \frac{F_t}{y} = 6737 \text{ , N} \tag{3}$$

Verification to the contact pressure in the chain articulation is met, thus the effective contact pressure p_s , in N/mm^2 (MPa), is smaller than the admitted pressure p_v,

$$p_s = \frac{F_a}{2\,A_a} = 23{,}844 \text{ , MPa} \qquad p_v = \frac{P_{vy}}{y} = 28{,}571 \text{ , MPa} \tag{4}$$

where: A_a is the surface of the chain articulation, A_a = 89 mm^2; P_{vy} – ratio of dynamic pressure and shock coefficient y, P_{vy} = 18.

Toothed wheel transmission or gear provides a compact design with a good reliability. This should be defined by reference rack, teeth inclination angle, and module, number of teeth, axial distance, profile movement and wheel generation module.

Based on the literature of speciality, MathCAD program has been devised, which allowed establishing the gear parameters, number of teeth z = 42, module m = 5 mm, distance between axes a = 210 mm, and based on those, geometrical wheel elements have been calculated, as well as verification of the gear at contact tension between the teeth sides and bending at the base of the teeth.

For an improvement steel allowing a hardness of the teeth sides of more than 300 HB, an admitted contact fatigue tension of 750 Mpa results, determined with the equation:

$$HB = 3000 \text{ MPa}, \quad \sigma_{H\,lim} = 0{,}15\,HB + 300 = 750 \text{ MPa.} \tag{5}$$

Effective admitted contact fatigue tension between the sides of the teeth is determined with the equation:

$$\sigma_H = Z_H\,Z_E\,Z_\varepsilon\,Z_\beta\,\sqrt{\frac{F_{tH}\,K_A\,K_v\,K_{H\beta}\,K_{H\alpha}}{b_w\,d_1}\,\frac{i+1}{i}} = 480{,}226 \text{ MPa,} \tag{6}$$

where: Z_H is the sides shape influence, Z_H = 2,49; Z_E – materials factor, Z_E = 189; Z_ε – minimum contact length influence factor, Z_ε = 0,871; Z_β – factor of use, K_A = 1,25; K_v – dynamic factor, K_v = 1,1; $K_{H\beta}$ – stress repartition factor over the width of the teeth, $K_{H\beta}$ = 1,1; $K_{H\alpha}$ – stress repartition factor between the teeth, $K_{H\alpha}$ = 1,32; b_w – teeth width, b_w = 30 mm; d_1 – drive wheel division diameter, d_1 = 210 mm; i –transmission ratio, i = 1.

The influence factor of the functioning duration on the contact stress is determined with the equation:

$$Z = Z_L\,Z_v\,Z_R\,Z_W\,Z_X = 0{,}866 \tag{7}$$

where: Z_L is the greasing influence factor, Z_L = 1,1; Z_v – peripheral speed influence factor, Z_v = 0,91; Z_R – teeth sides roughness influence, R_a = 1,6 μm, Z_R = 0,87; Z_W – sides roughness ratio factor, Z_W = 1; Z_X – dimension factor, Z_X = 1.

Safety coefficient of contact stress between wheel teeth sides should be higher than 1,15, condition which is fulfilled,

$$S_H = \frac{\sigma_{H\,lim}\, Z}{\sigma_H} = 1{,}35 \tag{8}$$

Admitted tension for pulsating bending fatigue at the base of the teeth is determined with the equation:

$$\sigma_{0\lim} = 385 + 0{,}057\ HB = 556\ \text{MPa.} \tag{9}$$

Effective tension for pulsating bending fatigue at the base of the teeth is determined with the equation:

$$\sigma_F = \frac{F_{tH}\ K_A\ K_v\ K_{F\beta}\ K_{F\alpha}}{b\ m}\ Y_{Fa}\ Y_{Sa}\ Y_\varepsilon\ Y_\beta = 82{,}82\ \text{MPa,} \tag{10}$$

where: $K_{F\beta}$ is the stress repartition factor along the width of the teeth for the stress at the base of the teeth, $K_{F\beta}$ = 1,06; $K_{F\alpha}$ – stress repartition factor between the teeth for the stress at the base of the teeth, $K_{F\alpha}$ = 1,46; Y_{Fa} – shape factor of the teeth for the stress at the base of the teeth, Y_{Fa} = 2,4; Y_{Sa} – tension concentrator factor at the base of the teeth, Y_{Sa} = 1,65; Y_ε – cover degree factor for bending stress, Y_ε = 0,69; Y_β – teeth inclination factor for bending stress Y_β = 1.

Safety coefficient for pulsating bending stress at the base of the wheel teeth should be higher than 1,25, condition which is fulfilled.

$$S_F = \frac{\sigma_{0\,lim}\ Y_\delta\ Y_R\ Y_X}{\sigma_F} = 6{,}08 \tag{11}$$

where: Y_δ is the sensitiveness of the material at fatigue stress, Y_δ = 1,03; Y_R – roughness factor for bending stress, Y_R = 0,88; Y_X – dimension factor for bending stress, Y_X = 1.

Based on radial forces from chain transmission, by toothed wheels and drilling on a central disk, with the help of FEMAP soft, the analysis with finite elements of the drive shaft has been done.

To simplify the calculation model the shaft has been approximated to a bar of circular cross section, equal to the section of support pivots in gaskets.

This simplification allowed a more exact positioning of forces and supports on the model nodes, as it is shown in Fig. 5.

From the analysis with finite elements e the Von Mises equivalent tension resulted, with small values of down to 16,52 MPa

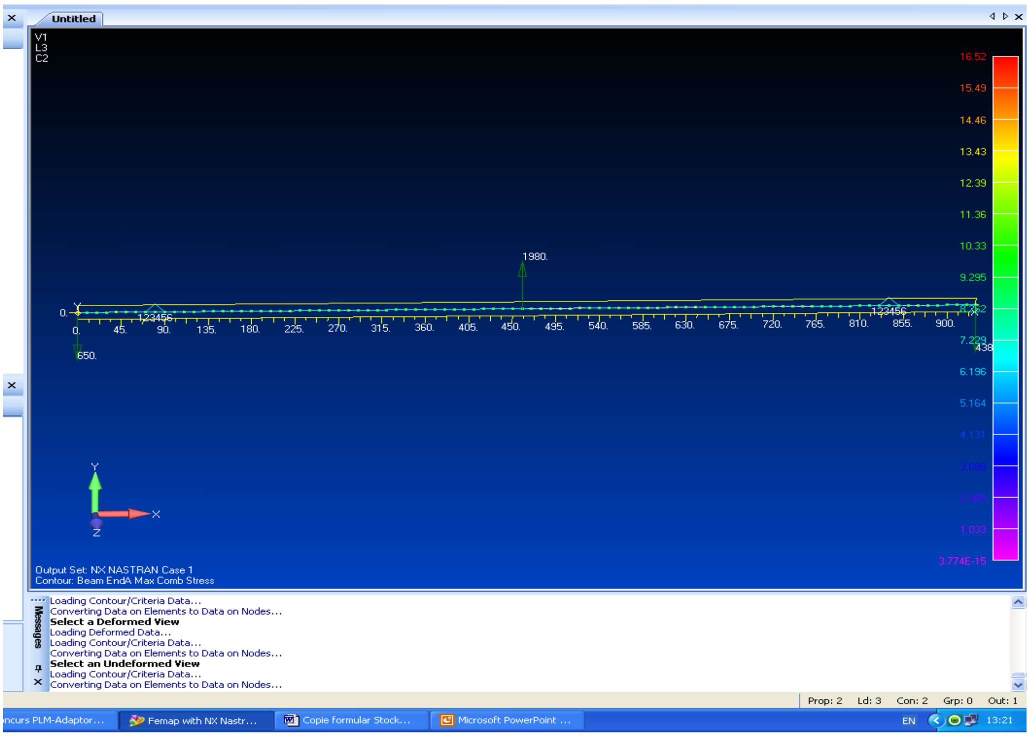

Fig. 5. Analysis with finite elements of the drive shaft

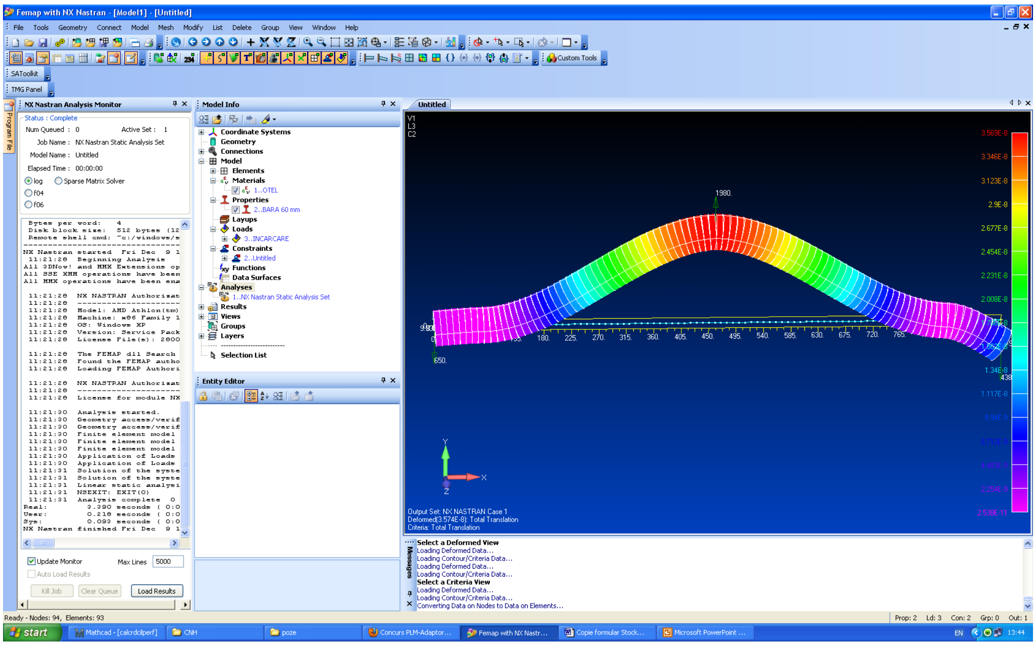

Fig. 6. Deformation module of the drive shaft

Fig 6 shows the deformation of the drive shaft under the action of loading forces, deformations are small, less than 0,001 mm, and do not influence the functioning of the drill.

4. CONCLUSIONS

By the use of the PET drill in waste processing, the conditions of PET packing and handling are improved and their deposit space is reduced.

With the help of NX7.5 soft, the set of PET drill has been modeled, with the possibility of rapid modification of the 3D model and execution documentation, function of the width of the press chute on which it is mounted.

The PET drill has a simple design, with rapid adapting possibilities to the technical exploitation methods, easy greasing of the chain and toothed wheel transmission.

Based on the dimensioning and verification calculations, it resulted that the functioning duration of the kinematic chain elements is of over 3000 hours, and by the modification of the chain transmission, from 12B double chain to 16A simple chain, it increases to over 10000 hours.

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