Jaliu Codruta, Saulescu Radu, Diaconescu
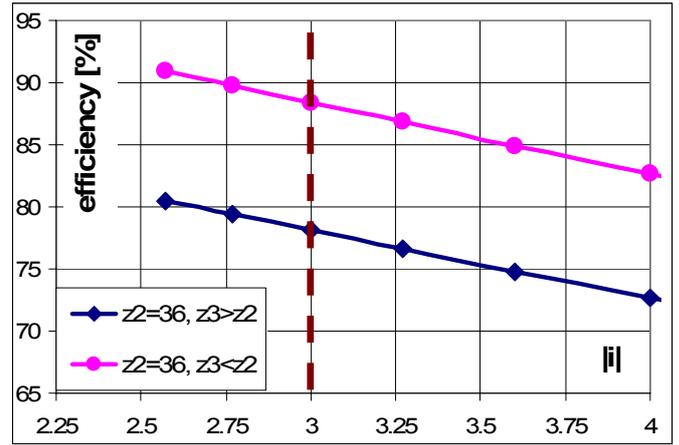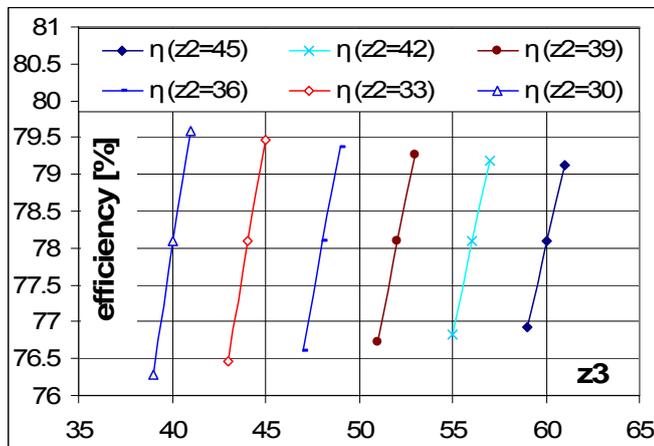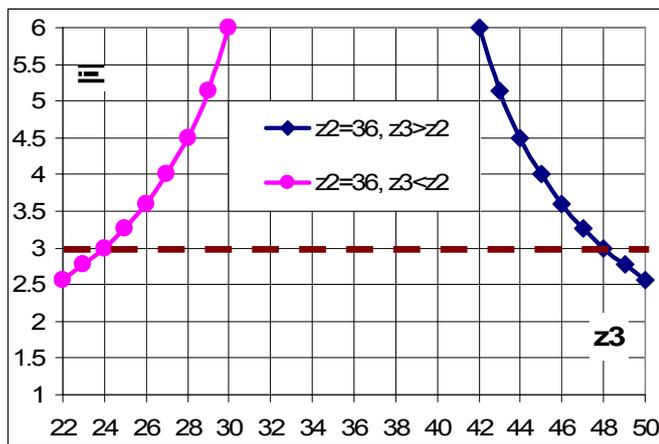Dorin, Neagoe Mircea, Climescu Oliver



a)



d)

Fig. 3. Numerical simulations for: d) the efficiency in the cases $z_3<z_2$ and $z_3>z_2$ for $z_2=36$.

The angular speeds and accelerations transmission functions (rel. 5) can be established based on relation (2), while the moments transmission function (rel. 6) based on relation (3), [4, 7]:

$$i_{1H}^3 = \frac{\omega_{13}}{\omega_{H3}} \Rightarrow$$

$$\omega_{H3} = \frac{\omega_{13}}{i_{1H}^3} = 3 \cdot \omega_{13}; \quad \varepsilon_{H3} = \frac{\varepsilon_{13}}{i_{1H}^3} = 3 \cdot \varepsilon_{13} \qquad (5)$$

$$\eta = \eta_{1H}^3 = \frac{-\omega_{H3}T_H}{\omega_{13}T_1} \Rightarrow T_H = T_1 \cdot i_{1H}^3 \cdot \eta_{1H}^3; \qquad (6)$$

in the case of neglecting friction, the moments transmission function is given by the following relation:

$$T_H = T_1 \cdot i_{1H}^3 = \frac{T_1}{3} = 0.3333 T_1$$

while in the case of considering friction, the function becomes:

$$T_H = T_1 \cdot i_{1H}^3 \cdot \eta_{1H}^3 = T_1 \cdot 0.3333 \cdot 0.781 = 0.2603 T_1$$



b)

## 3 Premises for Dynamic Modeling

The speed increaser dynamic modeling relies on the following premises:

- the dynamic model used in the paper is a simplified one, but with acceptable differences versus the real dynamic response [2, 6]; the considered accuracy is sufficient for this paper objective. The simplification consists in the following presumptions:
1) it is considered that the correlations generated by the planetary unit among the torques $T_1$, $T_H$ and $T_3$ (see Fig. 4,a) can be modeled in static conditions;



c)

Fig. 3. Numerical simulations for different combinations of teeth numbers of a) the multiplication ratio (multiplied 10 times), b) the efficiency, c) the multiplication ratio in the cases $z_3<z_2$ and $z_3>z_2$, for $z_2=36$ teeth.
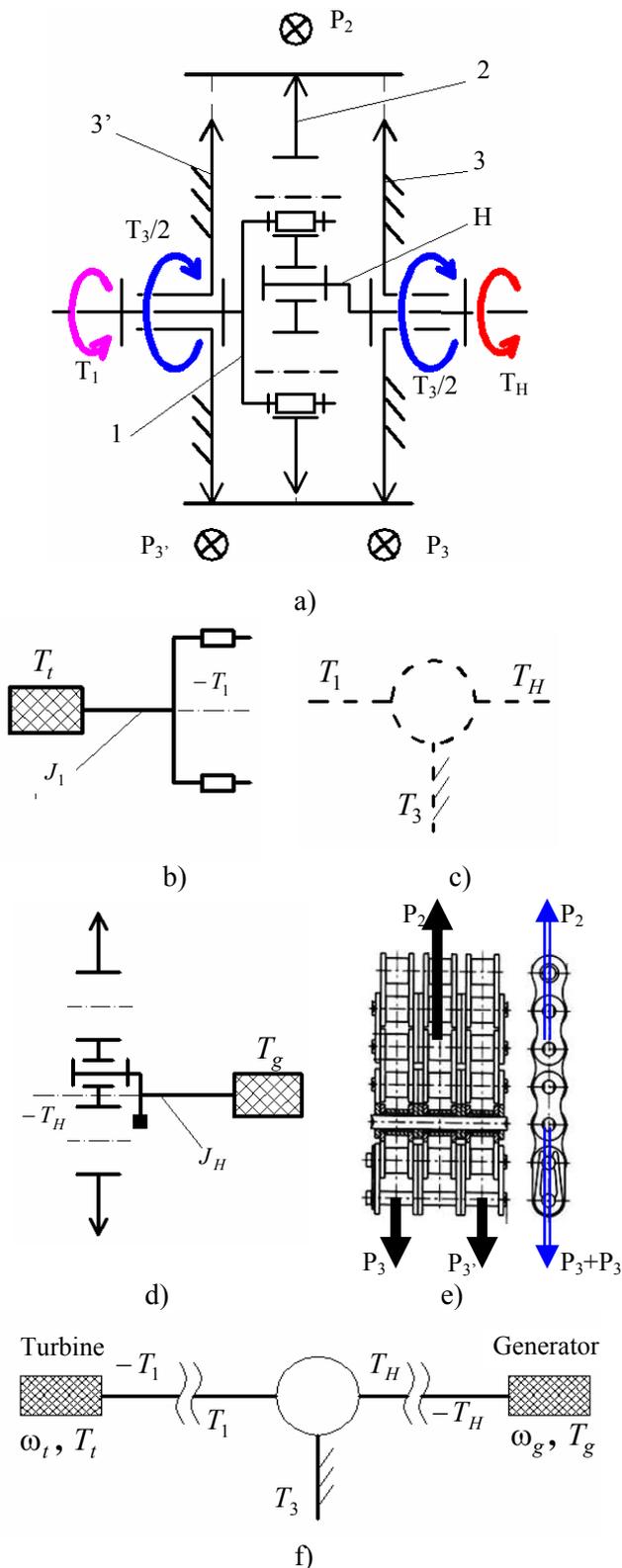
Fig. 4. The scheme of the chain planetary increaser: a) the external torques of the planetary unit and the chain forces, b) the dynamic scheme for the input element, c) the block diagram of the planetary speed increaser, d) the dynamic scheme for the output element, e) the chain forces' configuration, f) the block scheme of the considered machine (water turbine – speed increaser - electric generator).

2) the inertial moments of the satellite mass and, partially, of the chain are included in the mechanical inertia momentum of the output shaft (see Fig. 4,d);

3) the inertial effects of the central element 1 and, partially, of the chain are included in the mechanical inertia momentum of the input shaft (Fig. 4,b); thus, the inertial effects due to the satellite gear rotation are neglected (its mass being considered in the axial inertial moment of the afferent carrier shaft), while the inertial effects of the mobile central elements are considered integrated into the shafts that materialize the external links of the planetary gears; under this premise, the static correlations between the external torques of each planetary gear are valid (Fig. 4,a), while the dynamic correlations interfere only for the shafts that materialize the planetary gears external links. The mechanical inertia momentums of the two shafts (see Fig. 4) are:

$$J_1 = 0.035 \; ; \; J_H = 0.02 \; [Kgm^2] \tag{7}$$

- the rubbing effect is considered by means of the efficiency $\eta$;
- the turbine and generator are characterized by a power – speed characteristic and an efficiency – speed characteristic; in order to obtain the mechanical characteristics, the Turgo turbine and the generator were tested for different speeds on experimental stands (see Fig. 5); the results allow the establishment of the turbine and generator moment – speed characteristics, and, also, highlight the fact that the turbine efficiency is higher for lower speeds while the generator performances are better for higher speeds. The mechanical characteristics of the turbine and of the generator on their own shaft, used in the dynamic numerical simulations, have the following expressions (Fig. 6 and 7):



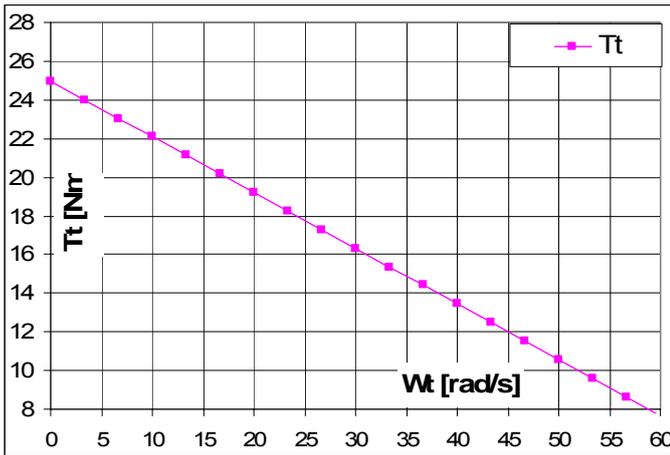Fig. 5. The generator parameters identification on experimental stand
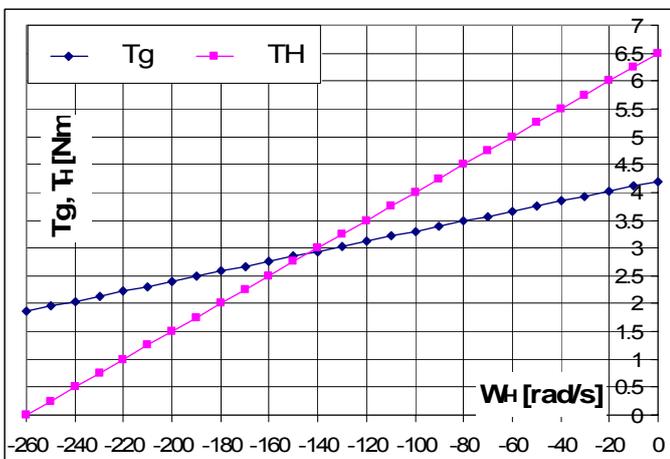
Fig. 6. The turbine mechanical characteristic



Fig. 7. The generator mechanical characteristic (Tg),
the turbine mechanical characteristic reduced on the
output shaft (-TH) and the stationary working point

$$T_t = -0.2881 \cdot \omega_b + 24.967 \ [\ Nm\ ] \qquad (8)$$
$$T_g = -0.009 \cdot \omega_g + 4.2 \ [\ Nm\ ] \ ;$$

- In order to establish the working point, the turbine mechanical characteristic is reduced to the speed increaser output shaft as $T_H = T_H(\omega_H)$; then, the two characteristics, of the turbine and generator are superposed (see Fig. 7).
- in the numerical simulations, the following values for the kinematical and dynamic parameters are considered:
  - the satellite and sun gears teeth numbers are $z_2 = 36$, $z_3 = 48$, for a module m = 3 mm,
  - the efficiencies of the pin coupling and chain transmission are $\eta_{12} = 0.995$, $\eta_{23} = 0.95$ [1, 4].

## 4  The Dynamic Model

In the dynamic modeling, the main objectives are to find the machine working point in stationary regime and the transmission functions for the angular speeds, accelerations and moments, relative to time:

$$\omega_t = \omega_t(t), \ \omega_{13} = \omega_{13}(t), \ \omega_{H3} = \omega_{H3}(t), \ \omega_g = \omega_g(t),$$
$$\varepsilon_t = \varepsilon_t(t), \ \varepsilon_{13} = \varepsilon_{13}(t), \ \varepsilon_{H3} = \varepsilon_{H3}(t), \ \varepsilon_g = \varepsilon_g(t), \qquad (9)$$
$$T_t = T_t(t), \ T_1 = T_1(t), \ T_H = T_H(t), \ T_g = T_g(t).$$

The validation of the dynamic model will be made by checking the concordance between the values of the speed and moment in stationary regime given by the dynamic model and the values of speed and moment corresponding to the working point.
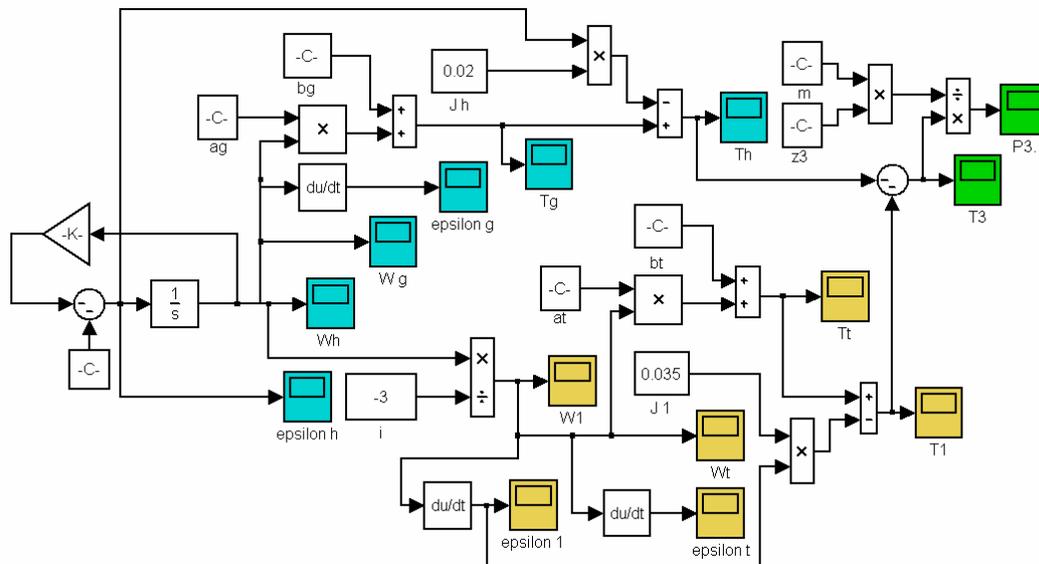


Fig. 8. The Simulink scheme that models the motion equation of the turbine-speed increaser-generator assembly

The working point in the stationary regime is given by the intersection of the turbine and generator characteristics (see Fig. 7) and is characterized by a speed $\omega_H = -143.6659$ [rad/s] and a moment $T_H = 2.9$ [Nm].

The dynamic modeling is made by means of Fig. 4, b,c, d, e and f, for the following cases:

• The motion equation is modeled by neglecting friction, and

• The motion equation is modeled by considering friction.

## 4.1 Case I, friction is neglected

In this case the Lagrange method is being used [7]:

$$\frac{d}{dt}\left(\frac{\partial E_c}{\partial \omega}\right) - \left(\frac{\partial E_c}{\partial \varphi}\right) = Q \tag{10}$$

According to Fig. 4, the kinetic energy $Ec$ and generalized force $Q$ have the following forms, in which $d\varphi/dt = d\varphi_H/dt = \omega_H$ represents the independent speed:

$$E_c \cong \frac{1}{2}\left(J_1 \cdot \omega_1^2 + J_H \cdot \omega_H^2\right);$$

$$Q.\omega_H = T_t \cdot \omega_1 + T_g.\omega_H \Rightarrow \tag{11}$$

$$Q = T_t \cdot i_{1H}^3 + T_g$$

in which $J_1$, represents the mechanical inertia momentum of the input element 1, respectively $J_H$, for the output element H (see Fig. 4).

Deriving $E_c$ relative to time, it is obtained:

$$\frac{\partial E_c}{\partial \omega_H} = \omega_1 J_1 . \frac{\partial \omega_1}{\partial \omega_H} + \omega_H J_H;$$

$$\frac{d}{dt}\left(\frac{\partial E_c}{\partial \omega_H}\right) = \varepsilon_1(1-i_0)J_1 + \varepsilon_H J_H; \tag{12}$$

$$\frac{d}{dt}\left(\frac{\partial E_c}{\partial \varphi}\right) = 0$$

From relations (11) and (12) it results:

$$\varepsilon_1(1-i_0)J_1 + \varepsilon_H J_H = T_t \cdot i_{1H}^3 + T_g \tag{13}$$

$$\varepsilon_1(1-i_0)J_1 + \varepsilon_H J_H = T_t(1-i_0) + T_g$$

$$\varepsilon_1 = i_{14}^3 \cdot \varepsilon_H = (1-i_0)\varepsilon_H$$

$$\varepsilon_H\left[J_1(1-i_0)^2 + J_H\right] = T_t(1-i_0) + T_g$$

$$T_t = a_t\omega_t + b_t, \; T_g = a_g\omega_g + b_g$$

$$\varepsilon_H\left[J_1(1-i_0)^2 + J_H\right] = (a_t\omega_t + b_t)(1-i_0) + a_g\omega_g + b_g$$

$$\varepsilon_H\left[J_H + J_1(1-i_0)^2\right] - a_t\omega_t(1-i_0) +$$
$$+ b_t(1-i_0) - a_g\omega_g - b_g = 0$$

For $\omega_t = \omega_1 = (1-i_0)\omega_H$ and $\omega_t = \omega_H$, the previous relation becomes:

$$\varepsilon_H\left[J_H + J_1(1-i_0)^2\right] - a_t\omega_t(1-i_0)^2 +$$
$$+ b_t(1-i_0) - a_g\omega_H - b_g = 0$$

By replacing the known parameters into relation (13), the dynamic equation when neglecting friction outcomes:

$$\varepsilon_H\left[J_H + J_1(1-i_0)^2\right] - \omega_H\left[a_t(1-i_0)^2 + a_g\right] - b_t(1-i_0) - b_g = 0 \tag{14}$$

where $a_t = -0.28818$ [Nms]; $b_t = 24.967$ [Nm];
$a_g = 0.009$ [Nms]; $b_g = 4.2$ [Nm];
$J_H = 0.02$ [$Kgm^2$]; $J_1 = 0.035$ [$Kgm^2$]
and $1 - i_0 = 0.3333$.

After numerical replacements, the following motion equation results:

$$\varepsilon_H \cdot 0.023889 + \omega_H \cdot 0.02302 + 4.12225 = 0 \tag{15}$$

## 4.1 Case II, friction is considered

In this case, according to Fig. 4,b, c, d and e, the following system of equations can be written using the Newton-Euler method:

$$T_1 + T_H + T_3 = 0; \tag{16}$$
$$T_1 \cdot i_{1H}^3 \cdot \eta_{1H}^3 + T_H = 0;$$
$$T_1 \cdot i_0 \cdot \eta_0 + T_3 = 0;$$
$$J_1\varepsilon_1 = T_m - T_1;$$
$$J_H\varepsilon_H = T_b - T_H.$$

The following relation is obtained through successive replacements:

$$\varepsilon_H\left[J_H + J_1(1-i_0)^2 \cdot \eta_{1H}^3\right] - T_t(1-i_0)\eta_{1H}^3 - T_g = 0$$

By taking into account friction, the equation used for modeling the machine dynamic system is obtained:

$$\varepsilon_H\left[J_H + J_1(1-i_0)^2 .\eta_{1H}^3\right] - \omega_H\left[a_t(1-i_0)^2 \eta_{1H}^3 + a_g\right] -$$
$$- t_b(1-i_0).\eta_{1H}^3 - b_g = 0. \tag{17}$$

After numerical replacements, the following motion equation results:

$$\varepsilon_H \cdot 0.023037 + \omega_H \cdot 0.01601 + 2.299677 = 0 \tag{18}$$

Jaliu Codruta, Saulescu Radu, Diaconescu
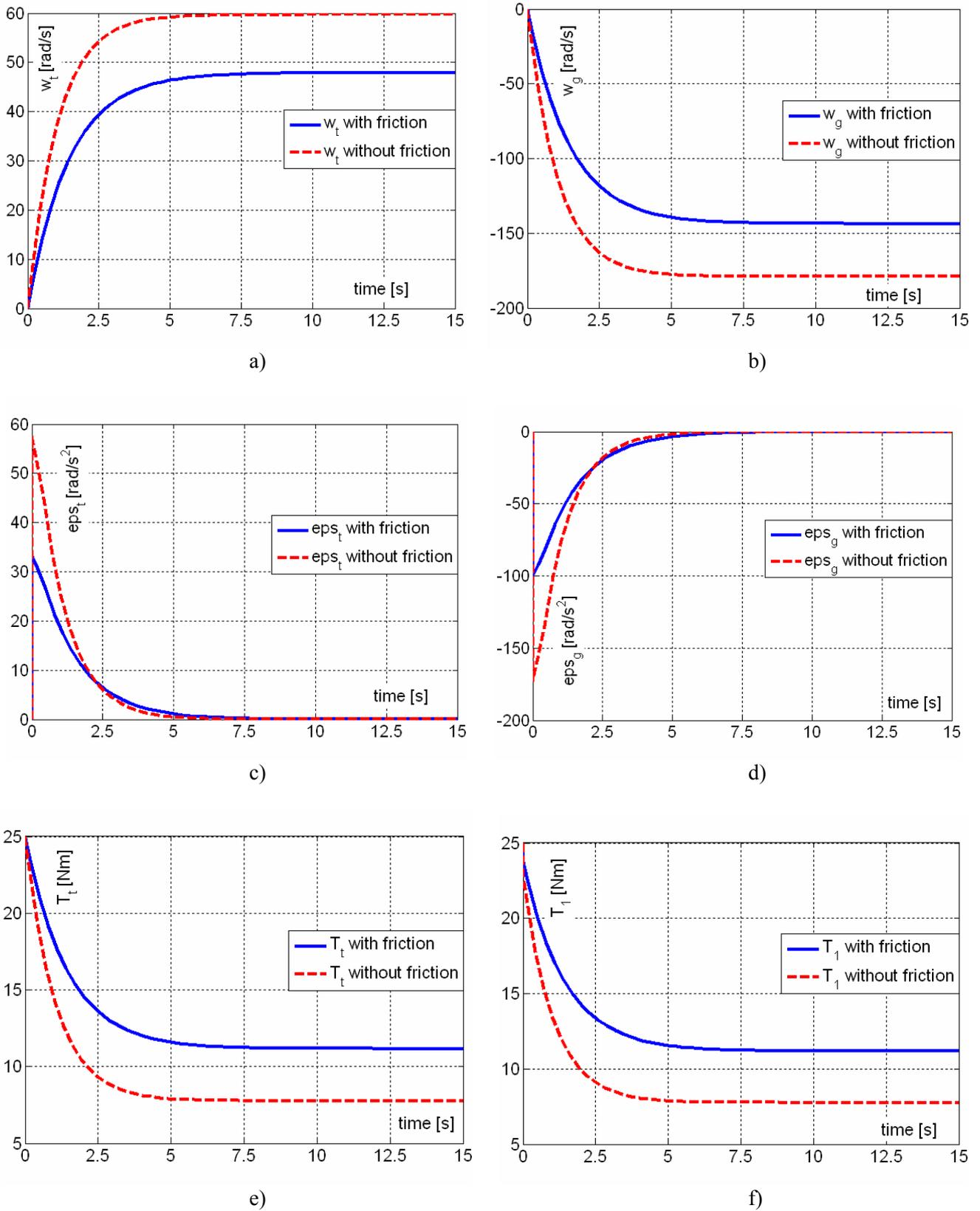Dorin, Neagoe Mircea, Climescu Oliver

Fig. 9. Dynamic response of the assembly: turbine angular speed (a), generator angular speed (b), turbine angular acceleration (c), generator angular acceleration (d), turbine moment (e), input moment (f)
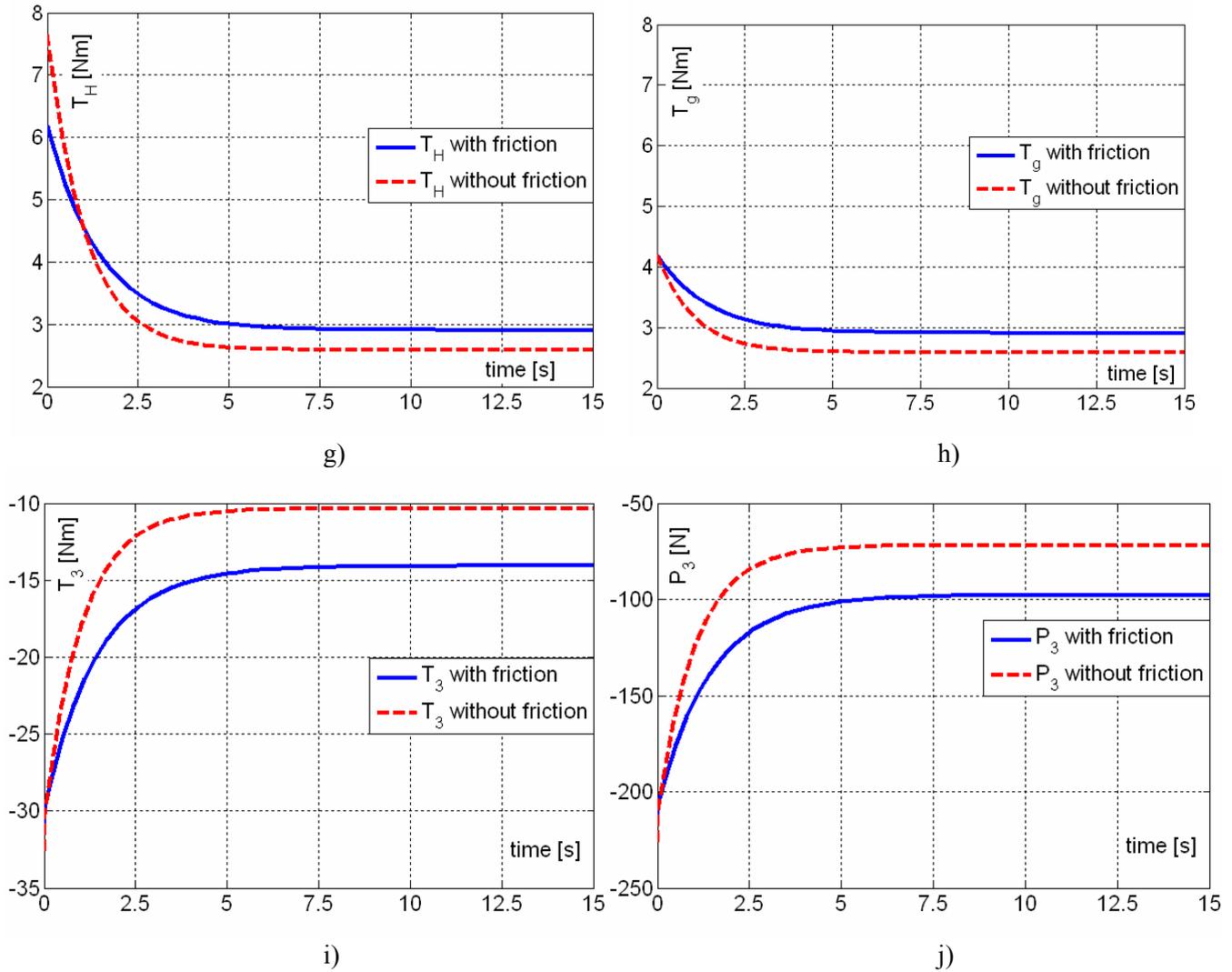
Fig. 9. Dynamic response of the assembly: output moment (g), generator moment (h), moment on the sun gear 3 (i), and force on chain (j).

Relation (18) represents the Turgo assembly motion equation in the case of considering friction. It can be easily observed that relation (15) is a particular case of relation (17).

## 5 Numerical Simulations

The values of the output and input angular speeds in stationary regime ($\varepsilon_H = 0$) are obtained from relations (15) and (18):

- when friction is neglected:

$\omega_H = -179.078$ ; $\omega_1 = 59.69191$, [rad/s]

- when friction is considered:

$\omega_H = -143.6659$ ; $\omega_1 = 47.88816$. [rad/s]

The Simulink scheme that models the motion equation of the turbine - planetary speed increaser – generator

assembly is presented in Fig. 8. The machine dynamic response, which consists of the speeds, accelerations and moments as functions of time, for the turbine, speed increaser and electric generator are depicted in the Fig. 9. The variations of the increaser input and output speeds and accelerations are not represented as they coincide to the speeds and accelerations of the turbine and generator, respectively. The forces that act on the chain, $P_2$, $P_3$ and $P_{3'}$ (Fig. 4,e) are given by the following relation, which is valid when neglecting the chain inertia:

$$P_3 \cong P_{3'} = \frac{T_3/2}{r_3} = \frac{T_3}{d_3} = \frac{T_3}{m_3 \cdot z_3} ;$$

where $m_3 = m = 3$ mm is the gears' module.

For the model validation, the working point values are compared to the simulation results; it can be observed that the values of speed and moment in the stationary

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