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COMPARISON STUDY OF TWO COMPETING MODELS OF AN ALL MECHANICAL POWER TRANSMISSION SYSTEM

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Keywords: Comparison, power transmission, simulation, Dymola

ABSTRACT

A comparison between two competing models of an all mechanical power transmission system is studied by using Dymola –software as the simulation tool. This tool is compared with Matlab/ Simulink –software by using functionality, user-friendliness and price as comparison criteria. In this research we assume that the torque is balanceable and transmission ratios are calculated. Using kinematic connection sketches of the two transmission models, simulation models are built into the Dymola simulation environment. Models of transmission systems are modified according to simulation results to achieve a continuous variable transmission ratio. Simulation results are compared between the two transmission systems. The main features of Dymola and MATLAB/ Simulink are compared. Advantages and disadvantages of the two softwares are analyzed and compared.

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1 INTRODUCTION

Conventional mechanical transmission systems have a constant transmission ratio from the start up. It is known that in principle a variable transmission ratio is advantageous for fuel consumption and for ride comfort. Two competing models of an all mechanical power transmission system are compared. The goal is to compare them by using two simulation programs and an analytic method. By using two unmatched planetary gears and one matched planetary gear continuous variable transmission ratio can be obtained [1]. The inner connections and interactions are not very clear for these two transmissions yet. So functions of the two transmission systems should be studied closely. Detailed research is also required to analyze the two competing models and compare them.

Dymola software for evaluation has been available for this research. In this research we will focus to build the dynamic models of the two mechanical power transmission systems. Dymola is suitable for modelling and simulating the dynamic behaviour of various kinds of physical objects. The most important feature of Dymola is the object-orientated formulation [2]. Simulink provides a block diagram interface that is built on the core MATLAB numeric, graphics, and programming functionality [3]. These applications are working in a different way. One of them will be purchased for future research. The comparison of these two softwares is also required to present a proposal for prospective users.

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2 INTRODUCTION TO TRANSMISSION SYSTEMS TO BE COMPARED

Two competing transmissions are studied, called A and B.

2.1 Introduction to Transmission system A

Transmission system A is an invention of power transmission using two sets of coupled unmatched planetary pinion gears of different size and a matched planetary gear. Its sketch map is shown in Fig.2.1.

The input shaft INSA is power source (engine or motor). The shaft drives planetary carrier C1A, which is housing three sets of coupled unmatched planetary pinion gears that are revolved within associated unmatched annulus gears. Planetary gears P2A, P3A are of different gear sizes and applied to Annulus gears. The two planetary P2A, P3A share a common planetary axle and intermesh with annulus gear A2A, A3A in sequence. Annulus gear A3A is connected to output shaft OUTSA as a solid one. A2A is connected to planetary sun gear S1A. Planetary carrier C2A also is connected to output shaft OUTSA as a solid one. The annulus gear A4A is fixed.

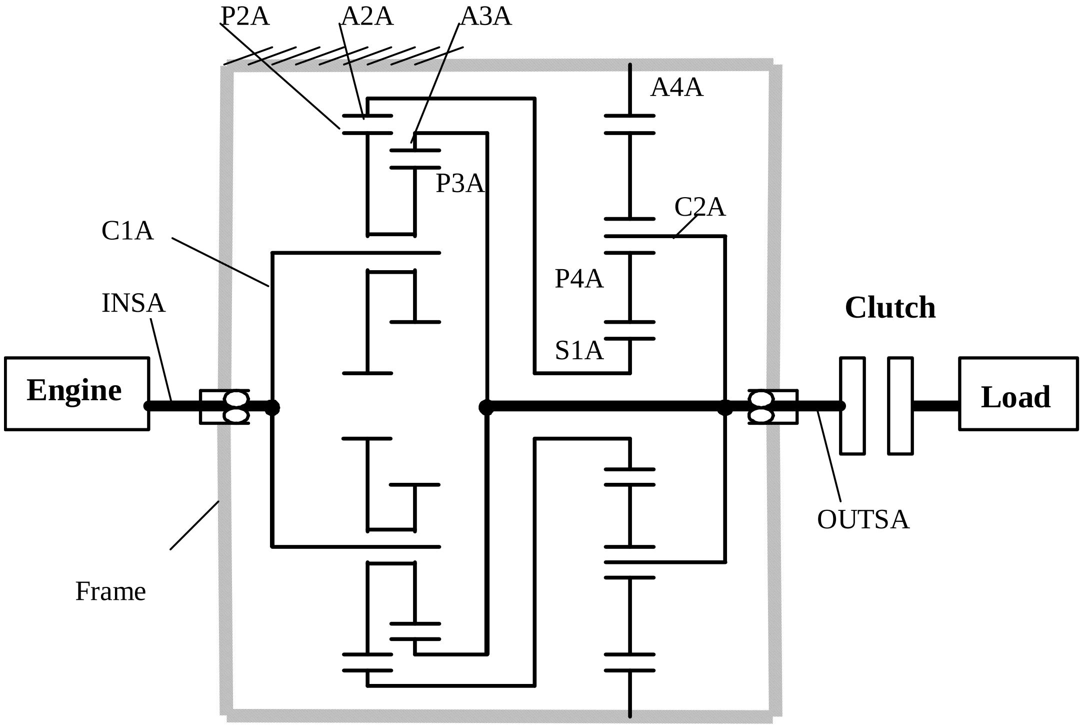


Figure 2.1 Sketch map of kinematics of Transmission System A

When input shaft INSA is rotated clockwise rapidly along with carrier C1A, for example at start-up, the unmatched but attached planetary pinion gears P2A and P3A revolve counter clockwise on their axles while inter-meshed with their associated annulus gears A2A and A3A. The difference between their sizes makes pinion gears P2A and P3A continually pull associated annulus gears A2A and A3A with them until the 1 to 1 ratio has been achieved. Gear A2A is rotated in clockwise to drive the matched planetary gear, which is connected with output shaft by planetary carrier C2A. Then A3A drives the output shaft directly. The power flows through two different routes. One route is INSA-C1A-P2A-A2A-S1A-P4A-C2A- OUTSA. The other route is INSA-C1A-P3A-A3A-OUTSA.

Fig.2.2 illustrates in detail the gear action, which takes place at start-up on three attached single pinion gears P2A and P3A. Suppose the teeth numbers of P2A and P3A are Z_{P2A} and Z_{P3A}; A2A and A3A are Z_{A2A} and Z_{A3A}. Now it is assumed that A2A is fixed. As planetary gears P2A and P3A are revolved clockwise as a unit, they revolve axially counter clockwise about their axles. Carrier C1A is revolved one full revolution on input shaft INSA while planetary gear P2A is revolved one full revolution from point p2aA to point p2aA due to annulus gear A2A is held stationary. If planetary gear P2A and P3A are not connected as a solid body, P3A will revolve from point p3aA to point p3bA according to the equation 2.1:

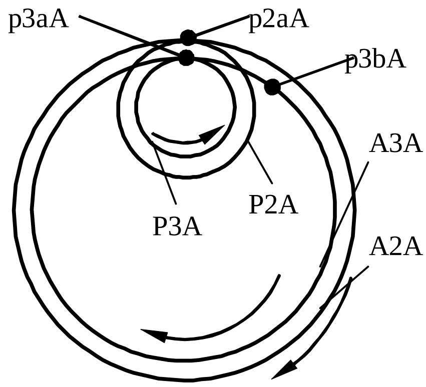


Figure 2.2 Action of Unmatched Planetary Gears

$$p3bA = 2\pi R_{A2A,P2A} / R_{A2A,P3A} \tag{2.1}$$

here :

$R_{A2A,P2A} = Z_{A2A} / Z_{P2A}; R_{A3A,P3A} = Z_{A3A} / Z_{P3A}$

Since Planetary gear P2A and P3A act as a solid ones, then P3A must also rotate one full revolution from p3aA to p3aA. Then annulus gear A3A must be pulled from p3bA back to p3aA. The rotation speed of annulus gears A3A can be calculated by the following equation:

$$\omega_{A3A} = \omega_{C1A}(1 - R_{A2A,P2A} / R_{A3A,P3A}) \tag{2.2}$$

Here :

ω_{C1A} – angular velocity of carrier C1A

Suppose the teeth number of planetary gear P2A and P3A is 15 and 10; A2A and A3A is 65, and 40. Then we can obtain the transmission ratio between carrier C1A and annulus gears A3A as:

$$R_{A3A,C1A} = \omega_{C1A} / \omega_{A3A} = 1/(1 - R_{A2A,P2A} / R_{A3A,P3A}) = -12$$

This is basically the key function of the unmatched planetary gear. It generates its torque, which is then applied to the output shaft OUTSA or sun gear S1A.

2.2 Introduction to Transmission system B

The transmission system B follows a US patent (No: 5713813) [1]. It is named Trans-Planetary Mechanical Torque Impeller. It consists mainly of two revolving and working units, one involving input and the other output. The assembly drawing of the transmission system B is shown in Fig.2.3. and the sketch map is presented in Fig.2.4.

Input shaft INSB is driven or revolved by a power source. Input shaft 1 is coupled to planetary carrier C1B, which houses two sets of unmatched pinion gears P1B and P2B and, which share a common planetary axle. Planetary pinion gears P1B are the larger primary pinion gears and they intermesh with primary annulus gear A1B. Planetary pinion gears P2B are the smaller secondary pinion gears and they mesh with the secondary annulus gear A2B. When input shaft INSB is rotated clockwise rapidly along with carrier C1B, for example at start-up, the unmatched but attached planetary pinion gears P1B and P2B revolve counter clockwise on their axles while inter-meshed with their associated annulus gears A1B and A2B. If all planetary gears were of equal sized, they would revolve aimlessly within their annulus gears resulting in no torque or rotation applied to the output section. The different sizes of them makes secondary the pinion gears P1B and P2B continually to pull the associated annulus gears A1B and A2B with them until the 1 to 1 ratio has been achieved. The greater the difference in diameters between the planetary pinion gears is, the smaller the input shaft to output shaft ratio becomes.

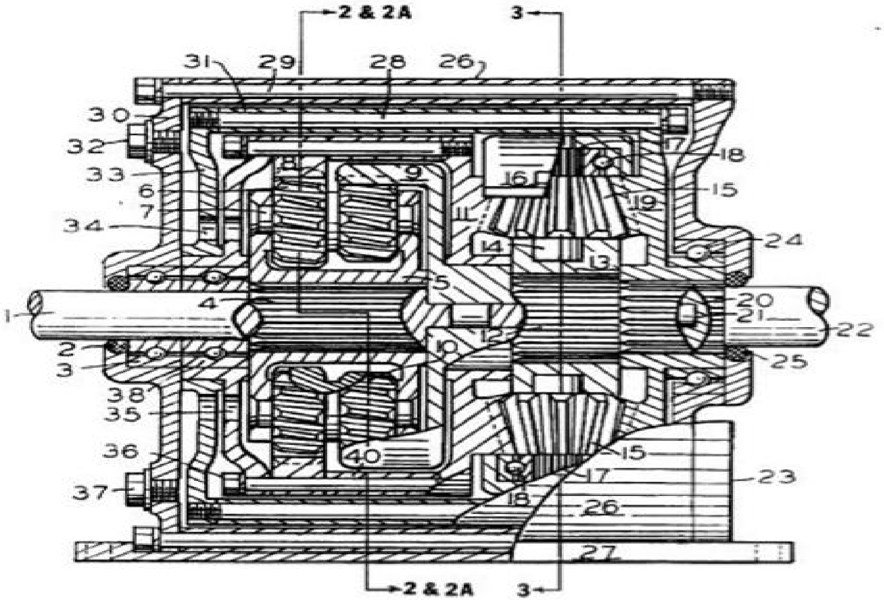


Figure 2.3 Assembly map of transmission B

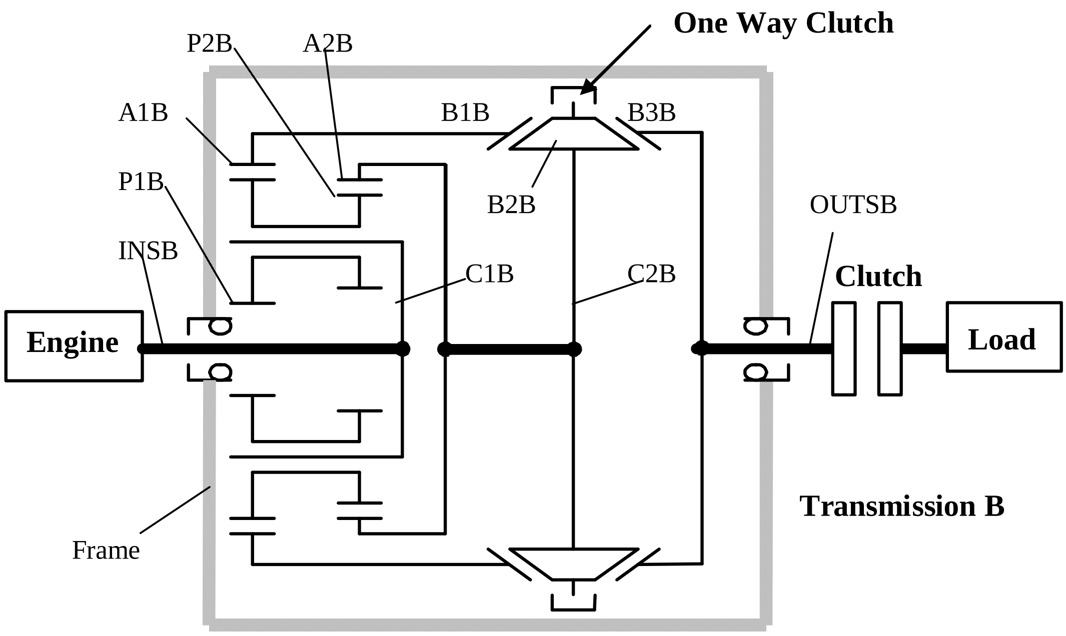


Figure 2.4 Sketch map of Transmission B

Annulus gear A1B is coupled to primary bevelled ring gear B1B stiffly, which in turn meshes with differential pinion gear B2B. Annulus gear A2B is coupled to differential drive carrier C2B as a solid body. Both annulus gears drive together a differential pinion gear B2B, which also meshes with output shaft bevelled ring gear B3B and, which then rotates the output shaft

OUTSB. Differential pinion gears B2B are supported and rotate on differential pinion gear bearings and are splined to ballbearing or ratchet type one way rotational clutches, which are housed in outer differential housings.

If primary beveled gear B1B and differential drive carrier C2B, along with annulus gears A1B and A2B were allowed to run free or over-run, then differential pinion gears B2B would run wildly around output shaft beveled ring gear B3B without any torque being applied to output shaft OUTSB. Ballbearing or ratchet type, one-way rotational clutches will allow differential pinion gears B2B to revolve on their bearings in one rotational direction only. For example, if differential pinion gears B2B were only limited to revolve counter-clockwise on their bearings, primary beveled ring gear B1B could, in effect, revolve counter-clockwise thereby making beveled ring gear B3B and output shaft OUTSB run clockwise. Beveled ring gear B1B can therefore move in a limited reverse direction, which adds to the forward rotation of output shaft OUTSB. However, it will never over-run or run faster than beveled ring gear B3B and output shaft OUTSB . It will, in effect, give annulus gear A1B a base for revolving primary pinion gears P1B and secondary pinion planetary gear P2B to revolve secondary annulus gear A2B and associated differential drive carrier C2B [1].

3 SIMULATION MODELS

3.1 Model of Transmission System A

Transmission A consists of two parts: The first is three unmatched planetary gears the second is a matched planetary gear. Fig 3.1 shows some details of unmatched planetary gears.

The first goal is to analyze the angular velocities of the unmatched planetary gear. According to the principle of kinematic continuity of displacements, the contact velocities of annulus gear and planetary gear are equal, so we can write according to equations 3.1 - 3.5:

$$v_A = v_P \tag{3.1}$$

$$v_A = r_A \cdot (\omega_A - \omega_C) \tag{3.2}$$

$$v_P = r_P \cdot (\omega_P - \omega_C) \tag{3.3}$$

$$z_A \cdot (\omega_A - \omega_C) = z_P \cdot (\omega_P - \omega_C) \tag{3.4}$$

$$R_{P,A} = \frac{z_A}{z_P} = \frac{\omega_P - \omega_C}{\omega_A - \omega_C} = \frac{\frac{\omega_P}{\omega_C} - 1}{\frac{\omega_A}{\omega_C} - 1} \tag{3.5}$$

Here:

r_A is annulus gear radius
r_P is planetary gear radius
z_A is teeth number of annulus gear
z_P is teeth number of planetary gear
$R_{P,A}$ is transmission ratio between annulus gear and planetary gear

The transmission ratios are now:

$$R_{P2A,A2A} = \frac{z_{A2A}}{z_{P2A}} = \frac{\omega_{P2A} - \omega_{C1A}}{\omega_{A2A} - \omega_{C1A}} = \frac{\frac{\omega_{P2A}}{\omega_{C1A}} - 1}{\frac{\omega_{A2A}}{\omega_{C1A}} - 1} \tag{3.6}$$

$$R_{P3A,A3A} = \frac{z_{A3A}}{z_{P3A}} = \frac{\omega_{P3A} - \omega_{C1A}}{\omega_{A3A} - \omega_{C1A}} = \frac{\frac{\omega_{P3A}}{\omega_{C1A}} - 1}{\frac{\omega_{A3A}}{\omega_{C1A}} - 1} \tag{3.7}$$

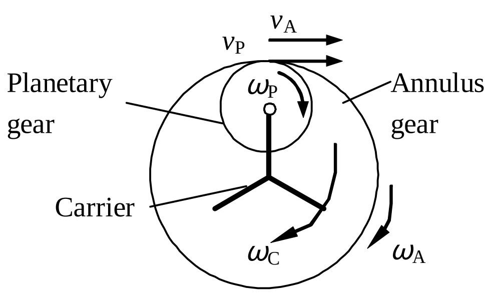


Figure 3.1 The action of unmatched planetary gear

Now the two gears P2A and P2A are stiffly connected to have the same speed

$$\omega_{P2A} = \omega_{P3A} \tag{3.8}$$

The carrier has the same speed as the input shaft:

$$\omega_{C1A} = 1 \bullet \omega_{INSA} \tag{3.9}$$

Here ωINSA is the angular velocities of in put shaft INSA, so:

$$\omega_{A2A} = \frac{\omega_{P2A} + \omega_{INSA}(R_{P2A,A2A} - 1)}{R_{P2A,A2A}} \tag{3.10}$$

$$\omega_{A3A} = \frac{\omega_{P2A} + \omega_{INSA}(R_{P3A,A3A} - 1)}{R_{P3A,A3A}} \tag{3.11}$$

Details of a matched planetary gear train are shown in Fig.3.2.

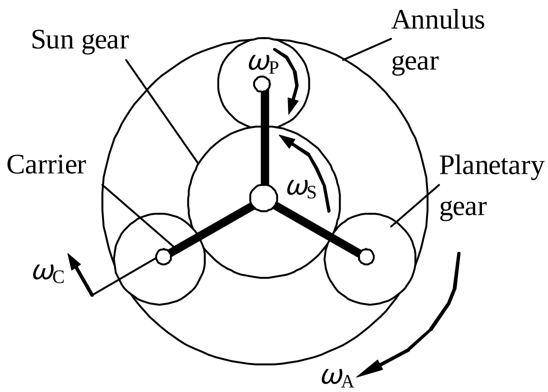


Figure 3.2 The action of matched planetary gear

The torque and power balances for the planetary gear train can be formulated by assuming that the rotations and torques all act in the same clock wise direction. The component powers can be expressed relative to the carrier velocity as presented in equations 3.12 – 3.16.

$$P_A = T_A \omega_A = T_A \omega_C + T_A(\omega_A - \omega_C) \tag{3.12}$$

$$P_S = T_S \omega_S = T_S \omega_C + T_S(\omega_S - \omega_C) \tag{3.13}$$

$$P_C = T_C \omega_C \tag{3.14}$$

$$T_A \omega_C + T_A(\omega_A - \omega_C) + T_S \omega_C + T_S(\omega_S - \omega_C) + T_C \omega_C = 0$$

$$T_A \omega_C + T_S \omega_C + T_C \omega_C = 0 \tag{3.15}$$

$$\frac{\omega_S - \omega_C}{\omega_A - \omega_C} = -\frac{T_A}{T_S} = -\frac{z_A}{z_S} = R_{S,A} \tag{3.16}$$

here:

T_A is the torque on annulus gear
T_S is the torque on sun gear
T_C is the torque on carrier
z_S is the teeth number of sun gear
$R_{S,A}$ is the transmission ratio between annulus gear and sun gear

For the transmission system A we have

$$\frac{\omega_{S1A} - \omega_{C2A}}{\omega_{A4A} - \omega_{C2A}} = -\frac{T_{A4A}}{T_{S1A}} = -\frac{z_{A4A}}{z_{S1A}} = -R_{S1A,A4A} \tag{3.17}$$

here:

T_{A4A} is the torque on annulus gear A4A
T_{S1A} is the torque on sun gear S1A
z_{A4A} is the teeth number of annulus gear A4A
z_{S1A} is the teeth number of sun gear S1A
$R_{S1A,A4A}$ is transmission ratio between annulus gear A4A and sun gear S1A
Now in this model A the annulus gear is fixed

$$\omega_{A4A} = 0 \tag{3.18}$$

Substituting this into equation 3.17 gives

$$\omega_{C2A} = \frac{\omega_{S1A}}{1 + R_{S1A,A4A}} \tag{3.19}$$

Sun gear S1A is stiffly connected to gear A2A

$$\omega_{A2A} = \omega_{S1A} \tag{3.20}$$

$$\omega_{C2A} = \frac{\omega_{P2A} + \omega_{INSA}(R_{P2A,A2A} - 1)}{R_{P2A,A2A}(1 + R_{S1A,A4A})} \tag{3.21}$$

Carrier C2A is stiffly connected to output shaft OUTSA,

$$\omega_{OUTSA} = \omega_{C2A} \tag{3.22}$$

Here ω_{OUTSA} is rotate speed of output shaft OUTSA.
Annulus gear A3A is also coupled output shaft OUTSA

$$\omega_{OUTSA} = \omega_{A3A} \tag{3.23}$$

Output speed is obtained as

$$\omega_{OUTSA} = \frac{\omega_{P2A} + \omega_{INSA}(R_{P2A,A2A} - 1)}{R_{P2A,A2A}(1 + R_{S1A,A4A})} \tag{3.24}$$

$$\omega_{OUTSA} = \frac{\omega_{P2A} + \omega_{INSA}(R_{P3A,A3A} - 1)}{R_{P3A,A3A}} \tag{3.25}$$

From equations 3.24 and 3.25 the total transmission R_A of transmission system A is obtained

$$R_A = \frac{\omega_{INSA}}{\omega_{OUTSA}} = \frac{R_{P2A,A2A} + R_{P2A,A2A}R_{S1A,A4A} - R_{P3A,A3A}}{R_{P2A,A2A} - R_{P3A,A3A}} \tag{3.26}$$

According to Fig.2.1 and the analyses of transmission ratio, simulation model was built by using Dymola software. The simulation model of transmission system A is shown in Fig.3.3. Components of the model and its function are listed in Table 3.1.

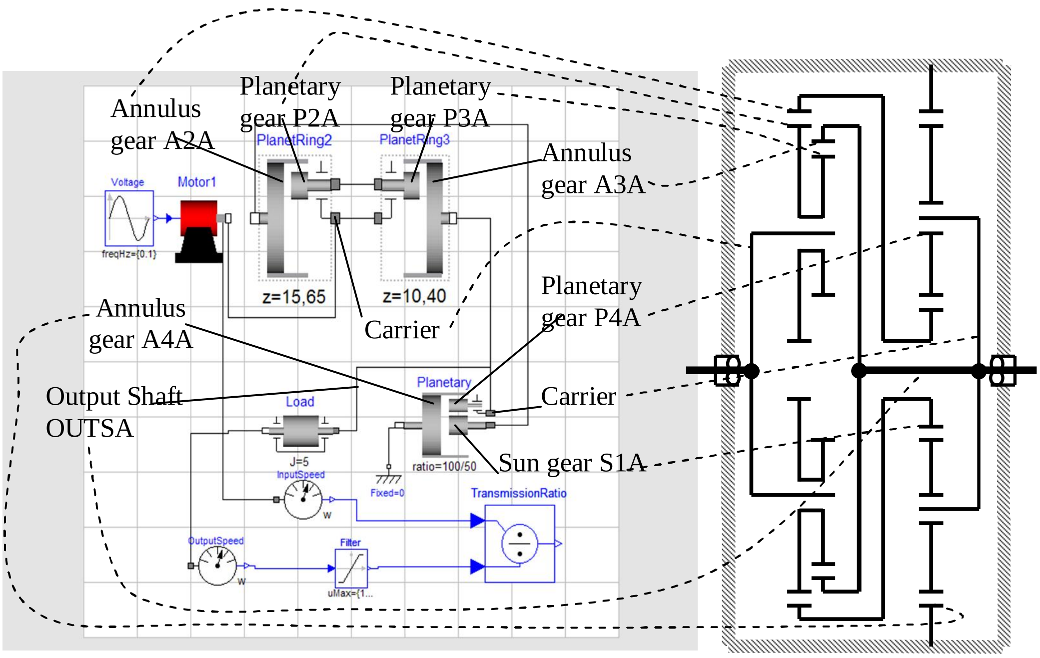
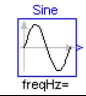
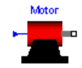
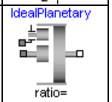
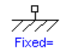
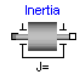
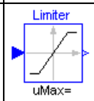
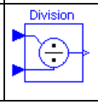


Figure 3.3 Dymola model of Transmission System A

More details of Planet Ring component may be found in particulars -menu. Fig.3.4 shows the details of Planet Ring component. Planet Ring Component belongs to Power Train -Library. This component has three main elements: planetary gear, ring gear and carrier. Planet rolling within a ring wheel and both wheels are connected by a carrier. Every element has its own port. The transmission is defined via teeth number.

Table 3.1 Details of Components

Component	Function	Roles in the model	Path in Dymola	ICON
sine	Generate sine signals	Voltage of DC Motor	Modelica.Blocks.Sources.Sine	
Motor	A basic model of an electrical dc motor	Power source of transmission	DriveLib.Motor	
PlanetRing	Planet and ring wheel of a planetary gearbox	Unmatched planetary gear	PowerTrain.Gears.PlanetRing	
IdealPlanetary	Ideal planetary gear box	Matched planetary gear	Modelica.Mechanics.Rotational.IdealPlanetary	
Fixed	Flange fixed in housing at a given angle	Fix annulus gear A4A	Modelica.Mechanics.Rotational.Fixed	
Inertia	1D-rotational component with inertia	Simulate Load	Modelica.Mechanics.Rotational.Inertia	
SpeedSensor	Ideal sensor to measure the absolute flange angular velocity	Measure input and output speed	Modelica.Mechanics.Rotational.Sensors.SpeedSensor	
Limiter	Limit the range of a signal	Limit output speed not equal to zero	Modelica.Blocks.Nonlinear.Limiter	
Division	Output first input divided by second input	Calculate transmission ratio	Modelica.Blocks.Math.Division	

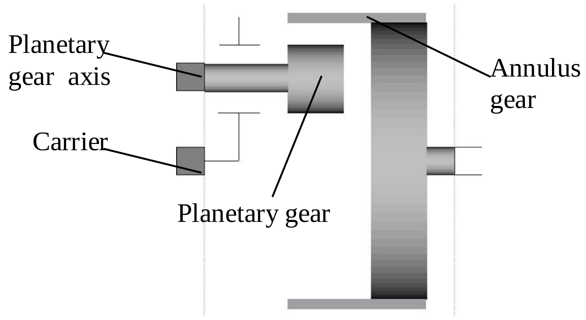


Figure 3.4 Planet Ring Component

Now there is no sun in P2A and the teeth numbers are for planet and annulus Z_{P2A}=15, Z_{A2A}=65. This component is used together with model "Planet Planet" to build up any type of planetary gearbox. So we can use this component to simulate the unmatched planetary gear. Power supply is a DC motor and its voltage is a sine signal. Two Planet Ring components are used to simulate the unmatched planetary gear. Planetary gears P2A and P3A share the same axis and carrier. This is expressed using the program object logic by connecting them by a line. P2A and P3A are connected by line. An Ideal Planetary component is used to simulate the matched planetary gear in transmission A. Annulus gear A2A is connected to sun gear S1A, Annuals gear A3A is connected to carrier C2A and output shaft OUTSA as shown in Fig.2.4. A "Fixed" component is used to fix annulus gear A4A. The load is an "Inertia" component and two "Speed Sensor" components are used to measure the input and output speed. So the total transmission ratio R_A can finally be calculated.

3.2 Model of Transmission System B

Transmission system B consists of two sets of unmatched pinion gears and a differential. Analogously with the analysis of unmatched planetary gear of the transmission system A, the equations of unmatched planetary of the transmission system B can be obtained as follows:

$$R_{P1B,A1B} = \frac{z_{A1B}}{z_{P1B}} = \frac{\omega_{P1B} - \omega_{C1B}}{\omega_{A1B} - \omega_{C1B}} = \frac{\frac{\omega_{P1B}}{\omega_{C1B}} - 1}{\frac{\omega_{A1B}}{\omega_{C1B}} - 1} \tag{3.27}$$

$$R_{P2B,A2B} = \frac{z_{A2B}}{z_{P2B}} = \frac{\omega_{P2B} - \omega_{C1B}}{\omega_{A2B} - \omega_{C1B}} = \frac{\frac{\omega_{P2B}}{\omega_{C1B}} - 1}{\frac{\omega_{A2B}}{\omega_{C1B}} - 1} \tag{3.28}$$

Here:

z_{A1B} is the teeth number of annulus gear A1B
z_{P1B} is the teeth number of planetary gear P1B
z_{A2B} is the teeth number of annulus gear A2B
z_{P2B} is the teeth number of planetary gear P2B
ω_{P1B} is the rotation speed of planetary gear P1B
ω_{A1B} is the rotation speed of annulus gear A1B
ω_{P2B} is the rotation speed of planetary gear P2B
ω_{A2B} is the rotation speed of annulus gear A2B

ω_{C1B} s the rotation speed of carrier C1B

Input shaft speed is the same to the speed of carrier C1B, and planetary gears P1B and P2B rotate as a solid one:

$$\omega_{C1B} = \omega_{INSB} \tag{3.29}$$

$$\omega_{P1B} = \omega_{P2B} \tag{2.30}$$

Here:
ω_{INSB} is the angular velocity of input shaft INSA.
From equation 3.27 - 3.30, speed of annulus gears A1B and A2B are obtained:

$$\omega_{A1B} = \frac{\omega_{P1B} + \omega_{INSB}(R_{P1B,A1B} - 1)}{R_{P1B,A1B}} \tag{3.31}$$

$$\omega_{A2B} = \frac{\omega_{P1B} + \omega_{INSB}(R_{P2B,A2B} - 1)}{R_{P2B,A2B}} \tag{3.32}$$

Some details of the differential gear are shown in Fig. 3.5:
The contact velocities of three gears of differential are equal:

$$v_{B1B} = v_{B2Bl}$$

$$r_{B1B}\omega_{B1B} = r_{C2B}\omega_{C2B} - r_{B2B}\omega_{B2B} \tag{3.33}$$

$$v_{B3B} = v_{B2Br}$$

$$r_{B3B}\omega_{B3B} = r_{C2B}\omega_{C2B} + r_{B2B}\omega_{B2B} \tag{3.34}$$

Here:
v_{B2Bl} is velocity of the left bevel gear B2B
v_{B2Br} is velocity of the right bevel gear B2B
r_{B1B} is rotation radius of bevel gear B1B
r_{B2B} is rotation radius of bevel gear B2B
r_{B3B} is rotation radius of bevel gear B3B
r_{C2B} is rotation radius of carrier C2B
Since the radii r_{B1B} , r_{B3B} , r_{C2B} are the same, then equations 3.33 and 3.34 can be transformed to

$$2\omega_{C2B} = \omega_{B1B} + \omega_{B3B} \tag{3.35}$$

$$2r_{B2B}\omega_{B2B} = r_{B1B}(\omega_{B3B} - \omega_{B1B}) \tag{3.36}$$

Annulus gear A1B and bevel gear B1B are solid one, annulus gear A2B is stiffly connected to carrier C2B. Thus the angular velocities are the same:

$$\omega_{B1B} = \omega_{A1B} \tag{3.37}$$

$$\omega_{C2B} = \omega_{A2B} \tag{3.38}$$

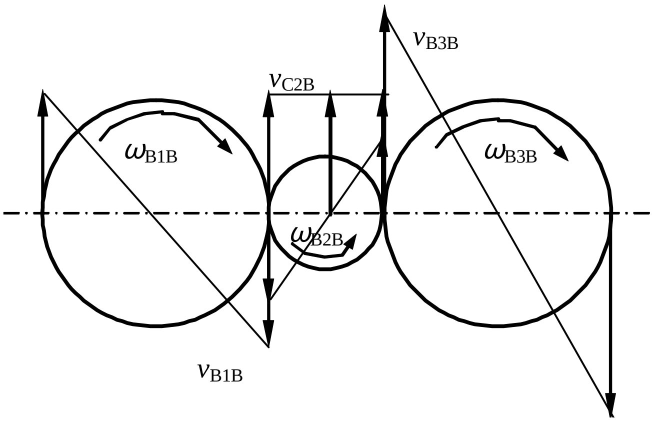


Figure 3.5 The function of differential gear

Bevel gear B3B is stiffly connected to the output shaft OUTSB

$$\omega_{OUTSB} = \omega_{B3B} \tag{3.39}$$

$$\omega_{OUTSB} = \omega_{INSB}(1 + \frac{1}{R_{P2B,A2B}} - \frac{2}{R_{P1B,A1B}}) + \omega_{P1B}(\frac{2}{R_{P1B,A1B}} - \frac{1}{R_{P2B,A2B}}) \tag{3.40}$$

That is the total transmission ration R_B of transmission B is:

$$R_B = \frac{1}{R_{1B} + R_{2B}R_{XB}} \tag{3.41}$$

Here:

$$R_{1B} = 1 + \frac{1}{R_{P2B,A2B}} - \frac{2}{R_{P1B,A1B}}$$

$$R_{2B} = \frac{2}{R_{P1B,A1B}} - \frac{1}{R_{P2B,A2B}}$$

$$R_{XB} = \frac{\omega_{P1B}}{\omega_{INSB}}$$

The transmission ratio R_B of transmission system B is not a constant. It changes with the input speed. Dymola simulation model can be made by using the previous model and the principles shown in Fig.2.3. The simulation model of transmission system B is shown in Fig.3.6. Most components and their functions are similar to those listed in table 3.1. Those two components, which were not used in the Dymola model for transmission system A are listed in table 3.2.

Table 3.2 Details of Components

Component	Function	Roles in the model	Path in Dymola	ICON
FreeWheel	Ideal free wheel	One way clutch of differential pinion gear	PowerTrain. Clutches. FreeWheel	
Differential	Differential gear box	Differential	PowerTrain.Gears. Differential	

Free wheel is defined as an ideal free wheel where the two flanges of the wheel can rotate freely with respect to each other as long as the relative angular velocity is positive. When the relative angular velocity becomes zero or negative, the two flanges are rigidly engaged. In other words, free wheel transfers a torque in one rotation direction only, which is the similar function compared with a diode, which transfers current only in one direction. We use a one way clutch to limit the differential pinion gear rotate to only in one direction.

The differential gear box splits the driving torque of an engine into equal parts for two driven output flanges. Modelica definition of this component is as follows:

```
model Differential "Differential gear box"
 parameter Real ratio=1 "gear ratio";
  Modelica.Mechanics.Rotational.Interfaces.Flange_a flange_engine
    "Flange of engine";
  Modelica.Mechanics.Rotational.Interfaces.Flange_b flange_left
    "Left (wheel) flange";
  Modelica.Mechanics.Rotational.Interfaces.Flange_b flange_right
    "right (wheel) flange";
equation
  flange_engine.phi = (flange_left.phi + flange_right.phi)*ratio/2;
  flange_left.tau = flange_right.tau;
  -ratio*flange_engine.tau = flange_left.tau + flange_right.tau;
end Differential;
```

From above we can find that since the equations of this component do not take care about the pinion gear, it cannot be used in simulation model directly. That's why we had to modify the Modelica definition as shown below:

```
model Differential "Differential gear box"
  parameter Real ratio=1 "gear ratio";
  Modelica.Mechanics.Rotational.Interfaces.Flange_a flange_engine
    "Flange of engine";
  Modelica.Mechanics.Rotational.Interfaces.Flange_a flange_pinion
    "pinion (gear) flange"
  Modelica.Mechanics.Rotational.Interfaces.Flange_b flange_left
    "Left (wheel) flange";
  Modelica.Mechanics.Rotational.Interfaces.Flange_b flange_right
    "right (wheel) flange";
equation
  flange_engine.phi - flange_pinion.phi = flange_left.phi;
  flange_engine.phi + flange_pinion.phi = flange_right.phi;
  flange_left.tau = flange_right.tau + flange_pinion.tau;
  flange_engine.tau = -flange_left.tau - flange_right.tau;
end Differential;
```

In this Modelica definition we need to insert a new port for pinion gear and rewrite the equations considering the effect of pinion gear. Then we can use this new component in the simulation model.

In the simulation model of the transmission system B, power is supplied by a DC motor and its voltage is a constant signal. This is the same as the model of modified transmission system A; two Planet Ring components are used to simulate the unmatched planetary gear. The planet gear axis and carrier of two Planet Ring components are connected according to Fig.2.3. Annulus gear A1B is connected to bevel gear B1B (left side of differential component) and annulus gear A2B is connected to carrier B1B (underside of Differential component). Bevel gear B3B (right side of differential component) is connected to output shaft OUTSB. Bevel gear B2B (the pinion gear of differential component) is connected to one way clutch (freewheel component). In order to limit bevel gear B2B to be able to rotate only in one direction, one port of Freewheel component is fixed. The load is also an inertia component and two Speed Sensor components are used to record the input and output speeds.

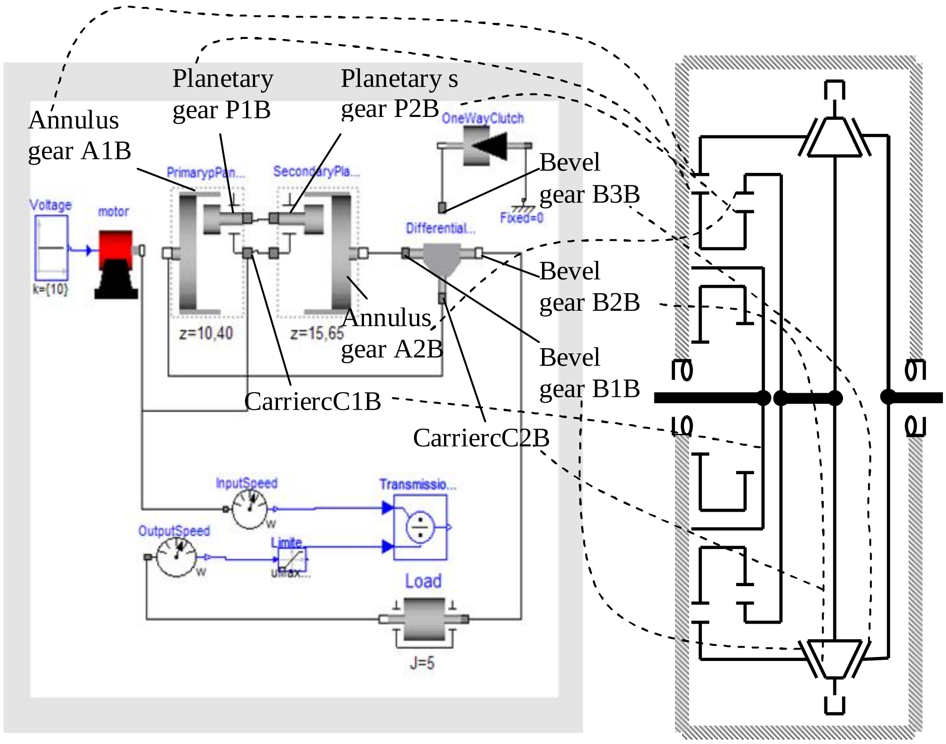


Figure 3.5 Dymola Model of Transmission System B

4 SIMULATION RESULTS

4.1 Simulation Results of Transmission System A

Dymola software was used to simulate behaviour of this transmission system A. The following simulation parameters are used:
Start Time is 0 sec., Stop Time is 5 sec., Number of Intervals is 500, Algorithm is Dassl and Tolerance is 0.0001. Input signal is a sine wave voltage whose amplitude is 10, frequency is 0.2Hz and offset is 12. The resistance of DC motor is 0.5Ohm, inductance is 0.05H, transformation coefficient is 1N.m/A and inertia is 0.001kg.m^2. Number of teeth of annulus A2A is 65, planet P2A is 15, annulus A3A is 40 and planet P3A is 10. Number of teeth of annulus A4A is 100 and that of S1A is 50.
Inertia of load is 5kg.m^2. Some simulation results are shown in Fig.4.1 to Fig.4.10

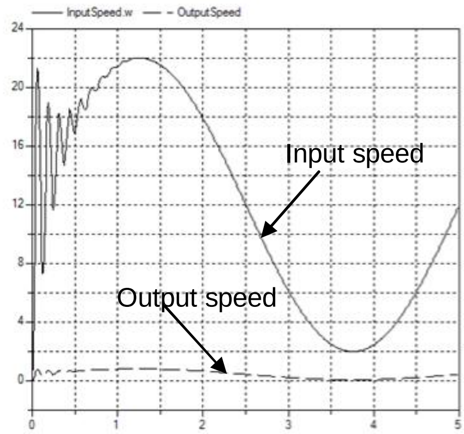


Figure 4.1 INSA and OUTSA speeda

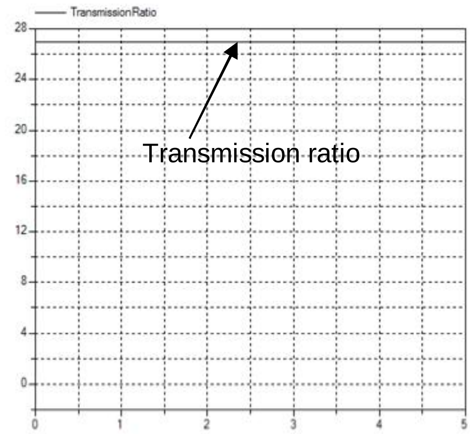


Figure 4.2 Transmission ratio of transmission system A

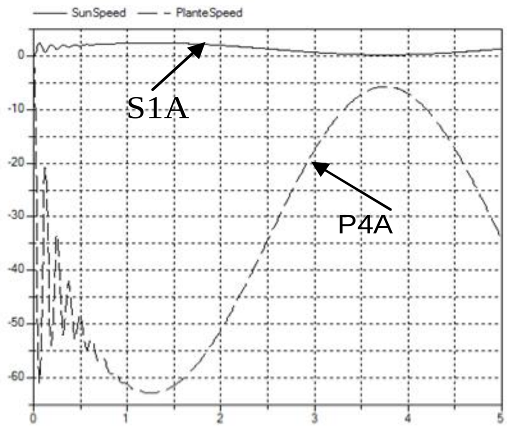


Figure 4.3 S1A and P4A speeds

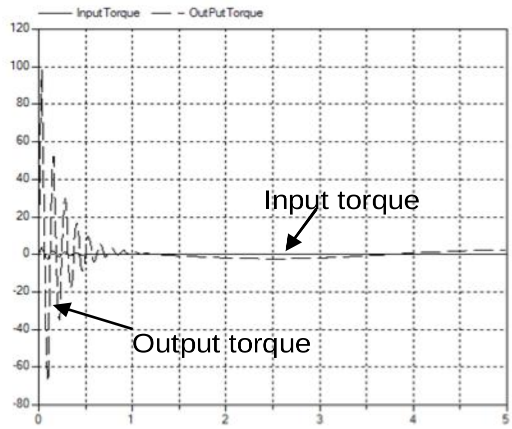


Figure 4.4 INSA and OUTSA torque

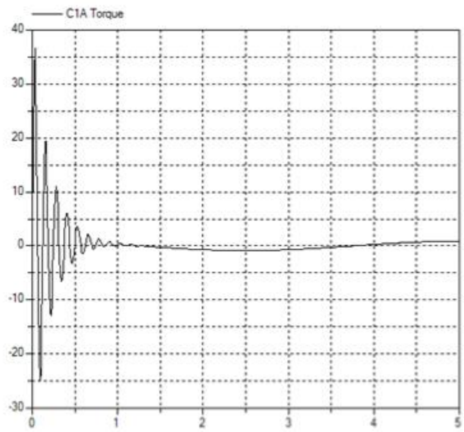


Figure 4.5 C1A torque

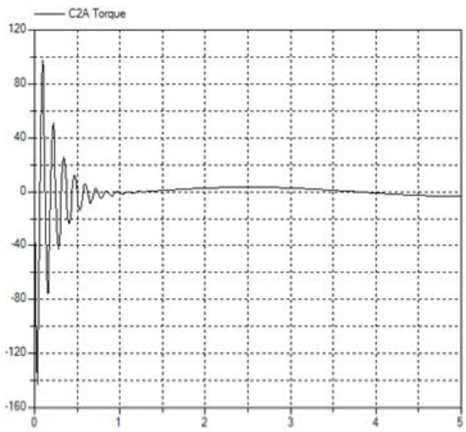


Figure 4.6 C2A torque

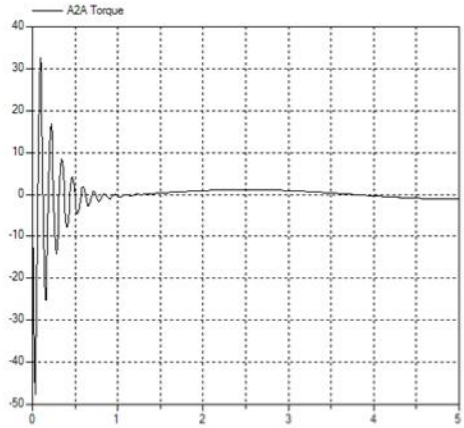


Figure 4.7 A2A torque

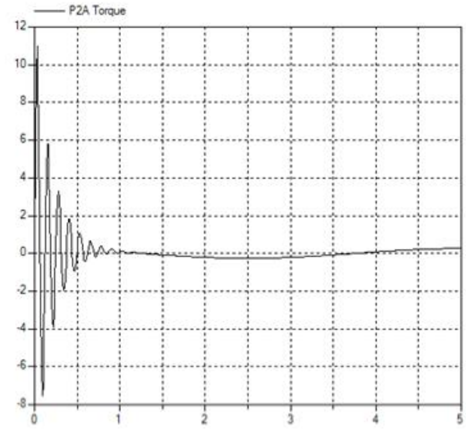


Figure 4.8 P2A torque

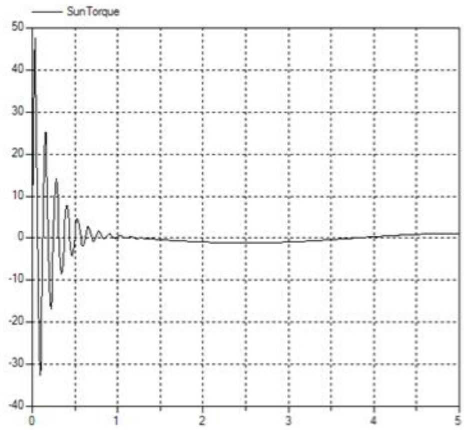


Figure 4.9 S1A torque

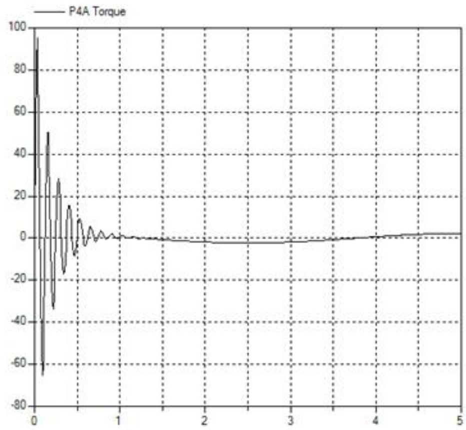


Figure 4.10 P4A torque

4.2 Simulation Results of Transmission System B

Let us use the same simulation tools as for transmission system A. Simulation parameters are also the same. Input signal is also a sine wave voltage whose amplitude is 10, frequency is 0.25Hz and offset is 12. In order to display full action of Transmission B, frequency is set to 2.25 Hz.
The parameter of DC motor are the same to that of Transmission A. Teeth numbers of A1B is 40 and P1B is 10, A2B is 65 and A3B is 15. The ratio of differential is 1. Inertia of load is 5kg.m^2.The patent does not describe the action of Transmission B very clear.

Some model assumptions have been made for analysis:
<u>Assumption 1</u>: The pinion gear B2B is limited to rotate in counter clockwise by one way clutch of transmission system B, see Fig.4.11(a). The simulation

results are shown as Fig.4.12 to Fig.4.15

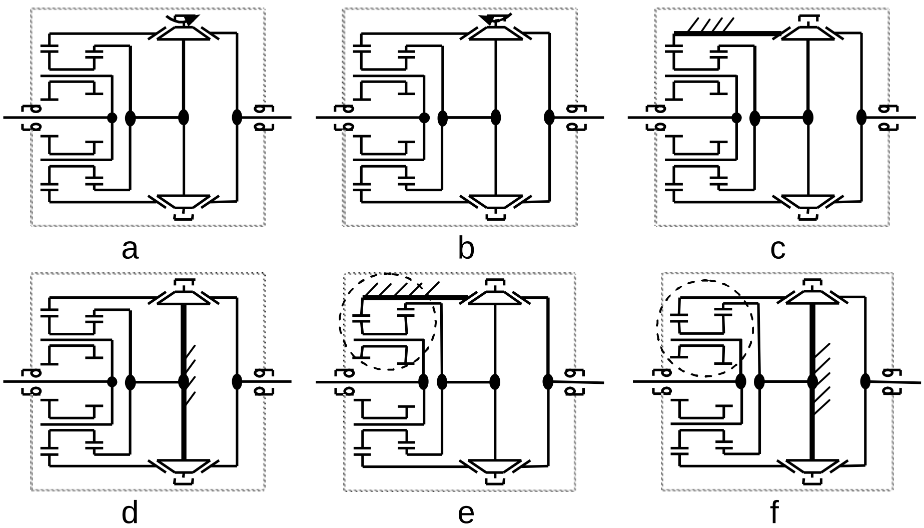


Figure 4.11. Hypothesis of transmission system B

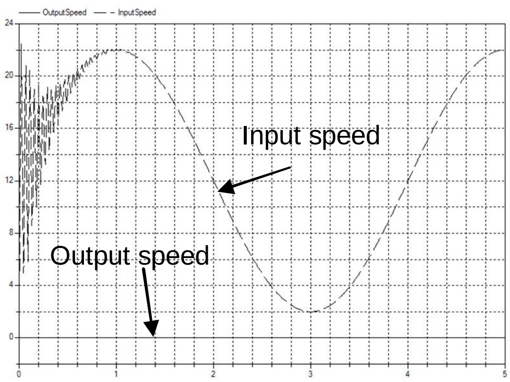


Figure 4.12 INSB and OUTSB Speed with one way clutch limiting counter clockwise rotation

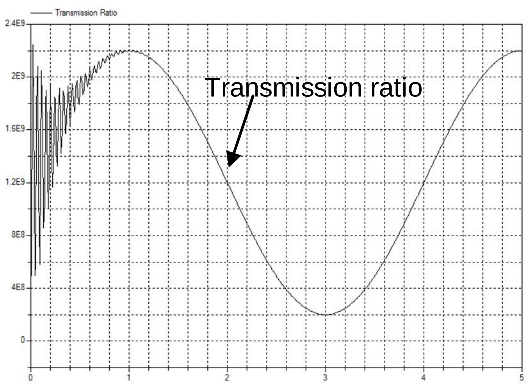


Figure 4.13 Ratio Transmission of system B with one way clutch limiting counter clockwise rotation

Assumption 2: The one way clutch of transmission B limited to run in clockwise, see Fig.4.11 (b). The simulation results are shown in Fig 4.16 to Fig 4.19.

Assumption 3: B1B (left side of differential) is fixed, see Fig.4.11(c). The simulation results are shown as Fig 4.20 and 4.21.

Assumption 4: C2B (underside of differential) is fixed, see Fig.4.11(d). The simulation results are shown in Fig 4.22 and 4.23.

Assumption 5: A1B is coupled with B1B and A2B is coupled with C2B and B1B is fixed, see Fig.4.11(e). The simulation results are shown in Fig 4.24 and 4.25

Assumption 6: A1B is coupled with B1B, A2B is coupled with C2B and C2B is fixed, see Fig.4.11(f). The simulation results are shown in Fig 4.26 and 4.27.

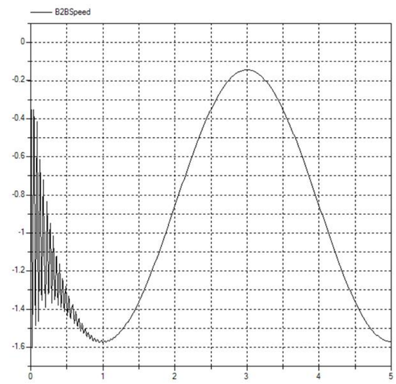


Figure 4.14 B2B Speed with one way clutch limiting counter clockwise rotation

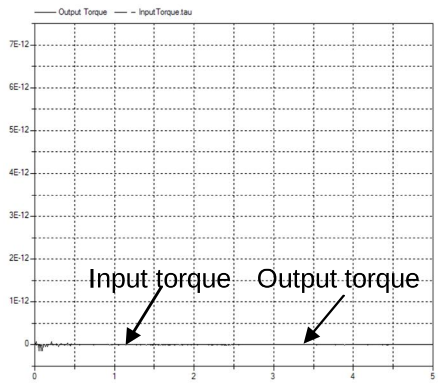


Figure 4.15 INSB and OUTSB Torques with one way clutch limiting counter clockwise rotation

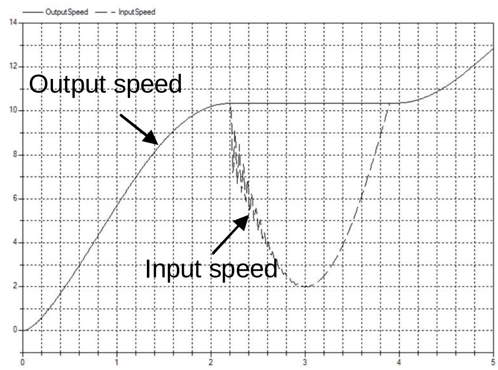


Figure 4.16 INSB and OUTSB Speed with one way clutch limited to clockwise

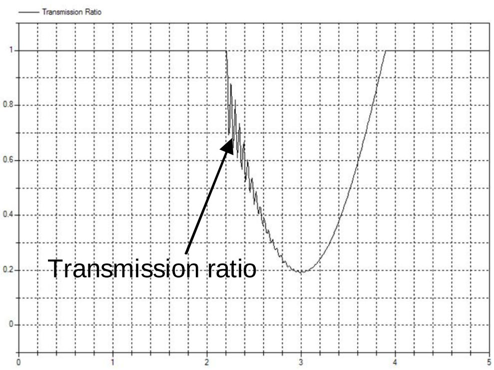
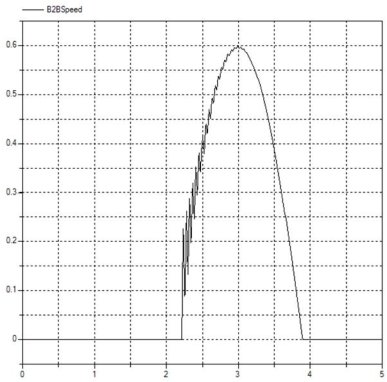


Figure 4.17 Transmission Ratio of system B with one way clutch limited to clockwise



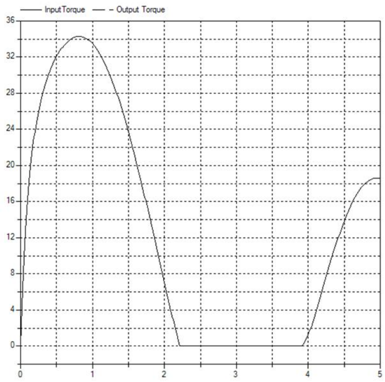
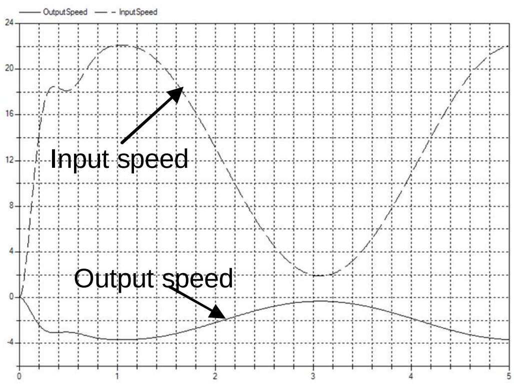


Figure 4.18 B2B Speed with one w ay clutch limited to clockwise

Figure 4.19 INSB and OUTSB torque with one way clutch limited to clockwise



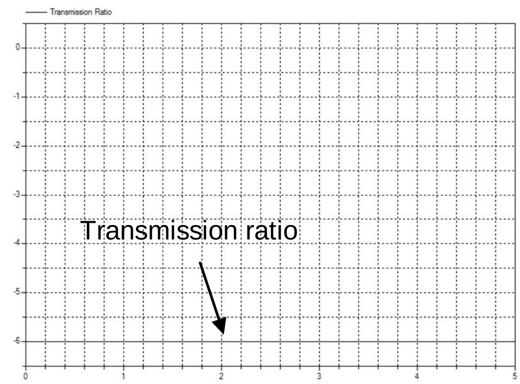
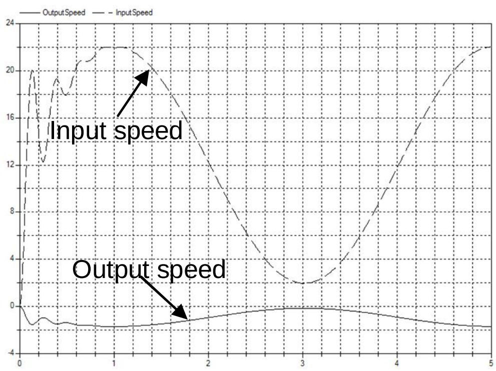


Figure 4.20 INSB and OUTSB Speed with B1B fixed

Figure 4.21 Transmission Ratio of system B with B1B fixed



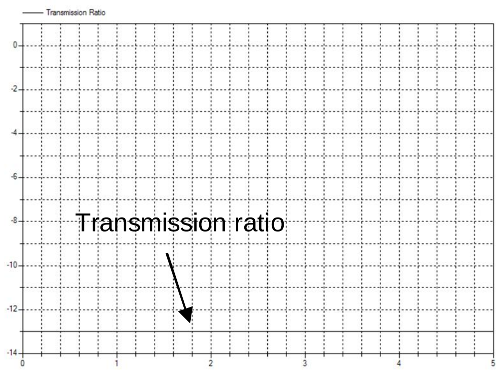


Figure 4.22 INSB and OUTSB speed with C2B fixed

Figure 4.23 Transmission ratio of system B with C2B fixed

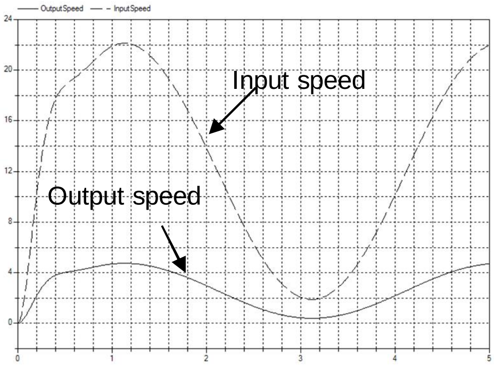


Figure 4.24 INSB and OUTSB fpeeds with B1B fixed after change the connection

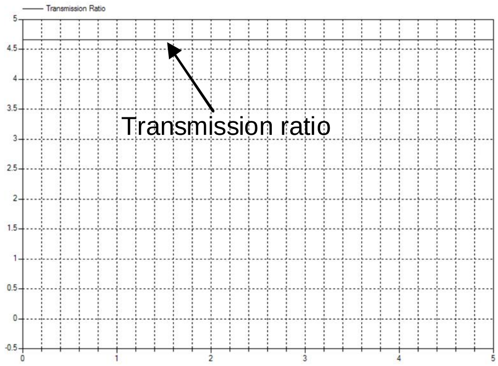


Figure 4.25 Transmission ratio of system B with B1B fixed after change the connection

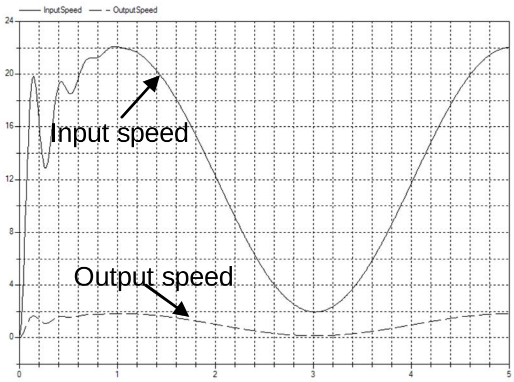


Figure 4.26 INSB and OUTSB speeds with C2B fixed after change the connection

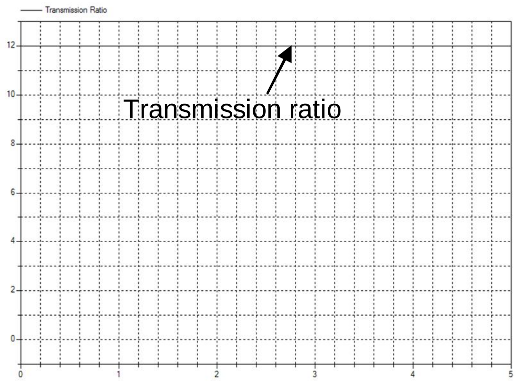


Figure 4.27 Transimission ratio of of system B with C2B fixed after change the connection

5 ANALYSIS OF SIMULATION RESULTS

5.1 Simulation Results of Transmission System A

When simulation is started the DC motor begins to accelerate by the sine wave voltage. The speed of DC motor is also a sine wave. There is some vibration at the moment of start up due to the response characteristic of a DC motor. The output torque reaches maximal value at the same time. This causes the load to accelerate quickly. Then acceleration gradually slows down and output torque tends to zero (see Fig 4.4). Torque values for some elements are shown in Fig 4.5 - 4.10. From Fig 4.1 and 4.2 it can be seen that the transmission ratio is 27, a constant one. This is the same as calculated from equation 3.28 by using the simulation component parameters. Thus the analysis and simulation model's results agree. Using this type of transmission we can get a wide transmission ratio with small actual size of the gear. But the transmission ratio is a constant one. However, the transmission ratio can not be changed automatically due to load changes.

5.2 Simulation Results of Transmission System B

When the one way clutch is limited to run counter clockwise all the elements in this transmission can run freely without constraint. So there is no output. Transmission ratio value tends to infinite value due to output speed is zero (see Fig.4.12 and 4.13). When pinion gear B2B is limited to run in clockwise by one way clutch all the elements in this transmission rotate as a solid body. As the DC motor begins to accelerate all the elements are fixed together because one way clutch can only be revolved in clockwise. Transmission ratio equals to 1 at this time. After acceleration output speed levels off and transmission ratio is one. Output speed keeps no change when DC motor speed begins to drop down. This is due to the fact that one way clutch can rotate only in clockwise. Transmission ratio's value changes continuously below one until input speed rises again. So transmission ratio can be continuously varied between 0 to 1 (see Fig.4.16 and 4.17). The bevel gear B1B is fixed with the speed of bevel gear B3B, which is double compared to that of carrier C2B. But C2B runs a reverse direction of DC motor due to the size difference between A1B and A2B. The transmission ratio is -6, which is a constant one (see Fig.4.20, 4.21). When carrier C2B is fixed with, the speed value of bevel gear B3B, it is equal to B1B, except the direction, which is the reverse. The transmission ratio value reaches -13, which is the negative maximal value (see Fig.4.22 and 4.23). When bevel gear B1B is still fixed but annulus gear A1B connects to B1B instead of C2B and annulus gear A2B connects to C2B instead of B1B, positive transmission ratios can be obtained. The absolute value of transmission ratio is smaller than shown Fig.4.20 (see Fig.4.24 and4.25). When bevel gear C2B is fixed, annulus gear A1B connects to B1B and annulus gear A2B connects to C2B, positive transmission ratio can be obtained. The absolute value of transmission ratio is smaller than that in Fig 4.21 (see Fig.4.26 and 4.27). The same results as listed above can be calculated by using equation 3.41. As a conclusion, a continuously variable ratio can be obtained using the one way clutch. But the value is too small to be useful in practical applications. Wide transmission ratio can be obtained by fixing certain elements where the value is a constant one. So according to authors' opinion, not only pinion gear B2B should be limited to rotate in only one direction but also carrier

C2B. In this case simulation results are shown in Fig.5.1 and 5.6.

From simulation results we can observe that the transmission ratio reaches a maximal value (although is negative) when DC motor is started (Fig.5.2). Output torque also reaches its maximum value (Fig.5.6). This enables the load to accelerate quickly (see Fig.5.1). At this moment carrier C2B is locked by One way clutch (Fig.5.3). After the vibration period the speed of DC motor tends to balance while input torque trends to zero. Carrier C2B is revolved in counter clockwise direction when DC motor slow down. Pinion gear B2B runs counter clockwise and output speed does not change. If the speed of DC motor is slower than output speed multiplied with the maximal transmission ratio, then differential pinion B2B gear will rotate in clockwise direction. Then one way clutch pinion B2B has to be stopped. The transmission ratio will be 1. The whole system rotates as a solid body. So transmission ratio can change continuously between -13 to 0. Positive transmission ratio can be obtained by using reverse gear.

Generally speaking, transmission system A can transmit torque with a certain ratio. Although large transmission ratios can be achieved with small practical size it is a constant one. In case of transmission system B, the transmission ratio will change continuously with load changes when carrier C2B is limited to rotate in only one direction. And further on, large wide ratio can also be obtained by using it twice. This means that transmission system B can be used more widely than the transmission system A.

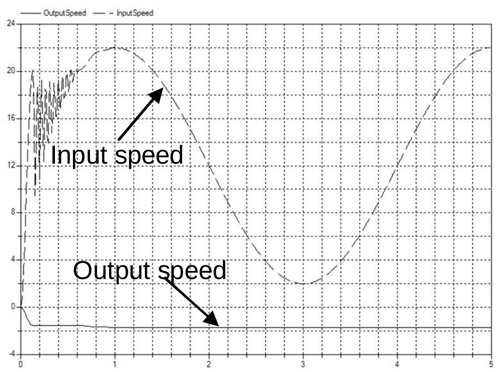


Figure 5.1 INSB and OUTSB speed with rotate direction of C2B is limited

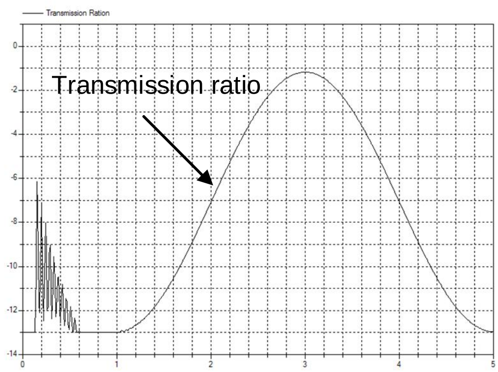


Figure 5.2 Transmission B ratio with rotate direction of C2B is limited

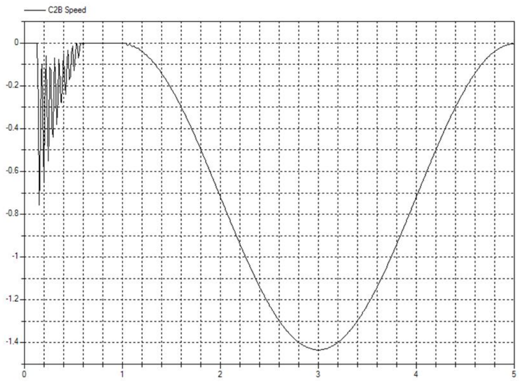


Figure 5.3 C2B speed with rotate direction of C2B is limited

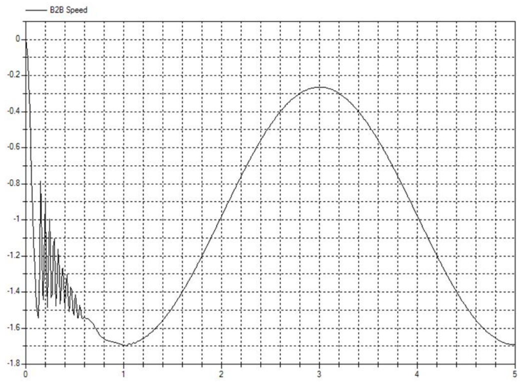


Figure 5.4 B2B Speed with rotate direction of C2B is limited

6 OVERVIEW OF DYMOLA SOFTWARE

6.1 Introduction

In engineering there is an increasing industrial interest in using simulation techniques [4]. Simulation is a fast and easy way to solve technical problems and to greatly reduce development cycle time and cut system and software design and prototype testing costs. Simulation tools can be divided into two types: one is block based modelling tools such as Matlab/ Simulink [3] and the other is object-oriented modelling tools such as Dymola [4].

In this study Dymola software is used to simulate dynamic performance of transmission systems. Dymola – Dynamic Modelling Laboratory – is suitable for modelling of various kinds of physical objects. It supports hierarchical model composition, libraries of truly reusable components, connectors and composite connections [2]. The most important feature of Dymola is that it uses object orientation and physics equations to build model. By this new modelling methodology automatic formula manipulation is used instead of manual conversion of equations to a block diagram. Due to this feature Dymola provides simpler modelling task to develop complex system models whose mathematic models are hard to formulate. In this research the actual functioning of transmission is not very clear. To develop mathematic models for each condition is waste of time. This is one of the main reasons why we have chosen Dymola software for modelling and simulating.

6.2 The logical Structure of Dymola

The structure map of the Dymola program is shown in Fig.6.1. Dymola software is an integrated environment for modelling and simulation. Models are composed using Modelica standard library, other open libraries, such as Mutibody System, commercial libraries, such as Power Train, and models developed by the user. User can develop models not only by connecting standard components from available libraries but user can also write his own equations. This is because Modelica supports both high level modelling by composition and detailed library component modelling by equations. User can use a graphical model editor to define a model by drawing a composition diagram by positioning icons that represent the models of the components, drawing connections and giving parameter values in dialogue boxes. The equation-based nature of Modelica is essential for enabling truly reusable libraries.

Measurement data and model parameters cover additional model aspects. Mass and inertia of 3D-mechanical bodies can be imported from CAD-packages. Visual properties may be imported in DXF- and STL-format. The icons of model components are defined either by drawing shapes in Dymola, or by importing graphics from other tools in bitmap format.

Dymola transforms a declarative, equation-based, model description into efficient simulation code. Advanced symbolic manipulation (computer algebra) is used to handle very large sets of equations. Dymola provides a self-contained simulation environment, but can also export code for simulation in Simulink. In addition to the usual offline simulation, Dymola can generate code for specialized Hardware-in-the-Loop (HIL) systems, such as, dSPACE, RTLAB, xPC and others. Experiments are controlled with a Modelica-based scripting language, which combines the expressive power of Modelica with access to external C libraries, e.g., LAPACK. The built-in plotting and animation features of Dymola provide the basis for visualization and analysis of simulation data. Experiments are documented with logs of all operations in HTML-format, including animations in VRML (Virtual Reality Modelling Language) and images. Dymola automatically generates HTML-documentation of models and libraries from the models themselves [5].

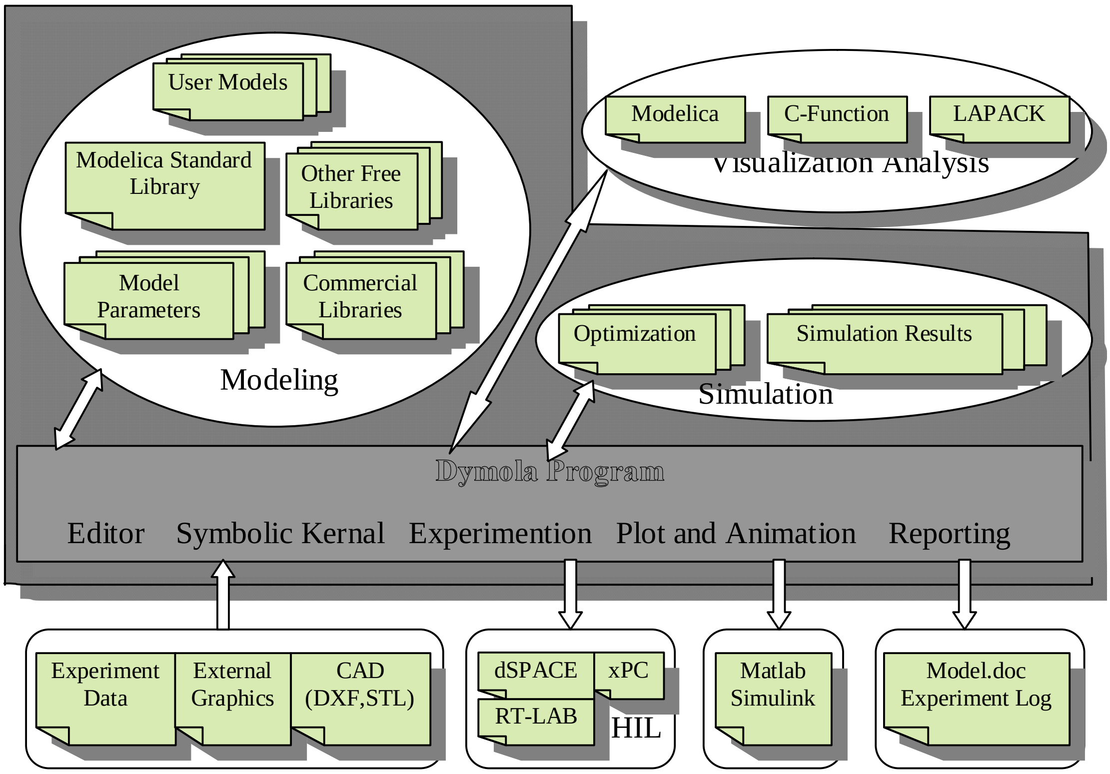


Figure 6.1 Structure Map of Dymola

6.3 Structure of the modelling environment

Modelling is the most important function of the simulation tool Dymola. Modelling environment plays an important role in modelling and simulation. The Dymola modelling environment can be divided into three major parts:

• The model editor. Using this, new models are composed. They can be existing components or made by equations written by the user. Default parameters are also set here.

• The Dymola main window is made to control simulations. Initial values are defined and simulation parameters are setup. Experiments can be setup interactively or through commands in script files.

• Simulation results are shown in animation and plot windows using animating components or plotting curve.

The flow chart of modelling and simulation is shown in Fig 6.2.

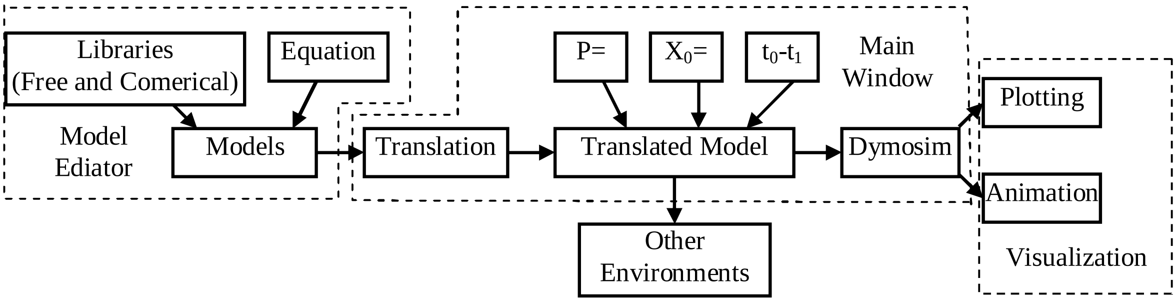


Figure 6.2 Flow Chart of Dymola modeling and Simulation

6.4 Application fields of Dymola software

Automotive area is the main application field for Dymola. Dymola has been used for several years within major automotive companies for complex simulations. Dymola provides a unique, efficient and integrated approach to the analysis of the multi-engineering aspects of vehicle design, which include acceleration, shift quality, fuel economy, emissions, vibrations, etc. In addition to automotive Dymola is also widely used in many other engineering domains such as aerospace, robotics, processes etc. Due to Dymola is an open modelling environment, it is possible to allow simulation of the dynamic behaviour and complex interactions between, for example, mechanical, electrical, thermodynamic, hydraulic and control systems.

6.5 System requirements for Dymola

Dymola can be used on platforms such as Widows, Linux and UNIX. Hardware requirements for Dymola on Linux and UNIX are not available while on Windows they are shown as below:

- PC/Windows 98/NT/2000/ME/XP
- > PIII 1.0GHz processor
- >128MB RAM
- >150MB Disk space

7 COMPARISON BETWEEN DYMOLA AND MATLAB/SIMULINK

7.1 Introduction

Dymola is a tool for modelling and simulation. MATLAB is a sophisticated language and a technical computing environment. It provides core mathematics and advanced graphical tools for data analysis, visualization, and algorithm and application development. Simulink is a simulation and prototyping environment for modelling, simulating, and analyzing real-world, dynamic systems. Simulink provides a block diagram interface that is built on the core MATLAB numeric, graphics, and programming functionality [3]. It is integrated with MATLAB, providing immediate access to an extensive range of tools for algorithm development, data visualization, data analysis and access, and numerical computation.

The relationship between MATLAB and Simulink is shown in Fig.7.1

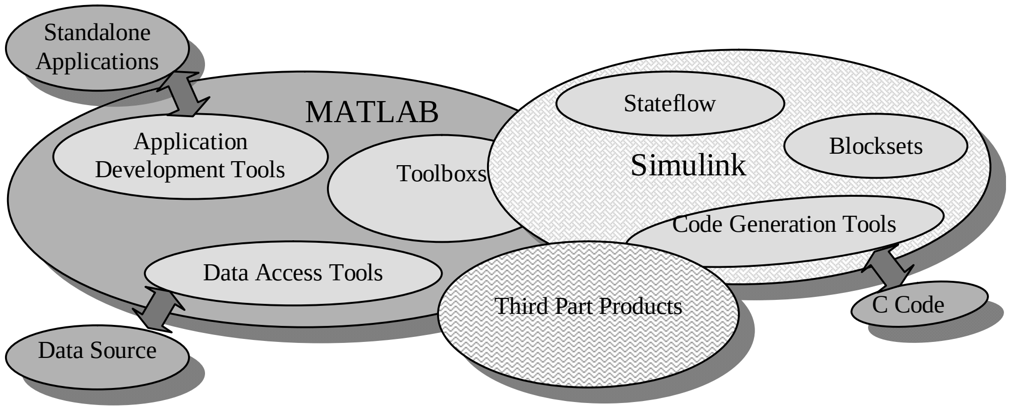


Figure 7.1 Relationship between Matlab and Simulink

Simulation functions consist of three parts: Stateflow, Blocksets and Code Generation tools. Stateflow is a good tool for modelling and designing event-driven systems. With Stateflow users can design the control or protocol logic quickly and easily. In Simulink it is assumed that all the object equations can be described by some basic blocks. Using these blocks users can build many different and complex models. Blocksets are made of many blocks in different fields including electrical power-system modelling, digital signal processing, fixed-point algorithm development, and more. These blocks are also developed by basic Simulink blocks and they can be incorporated directly into user's Simulink models. Code Generation Tools includes Real-Time Workshop and Stateflow. Coder can generate customizable C code directly from Simulink and Stateflow diagrams for rapid prototyping, hardware-in-the-loop simulations, and desktop rapid simulation. Of course some highly-optimized, application-specific functions are supported by Toolboxes to extend Simulink. Some features derived from above tools are presented of Simulink 6 in the following chapters.

7.2 Feature of Matlab/ Simulink.

In order to illustrate advantages of MATLAB/Simulink some features are listed below [6]:

■ Extensive and expandable libraries of predefined blocks

■ Interactive graphical editor for assembling and managing sophisticated block diagrams

■ Ability to manage complex designs by segmenting models into hierarchies of design components

■ Model Explorer to navigate, create, configure, and search all signals, parameters, and properties of user's model

■ Ability to interface with other simulation programs and incorporate user-written code, including MATLAB algorithms

■ Option to run fixed- or variable-step simulations of time varying systems interactively or through batch simulation

■ Functions for interactively defining inputs and viewing outputs to evaluate model behaviour

■ Graphical debugger to examine simulation results and to diagnose unexpected behaviour in user's design

■ Full access to MATLAB for analyzing and visualizing data, developing graphical user interfaces, and creating model data and parameters

■ Model analysis and diagnostics tools to ensure model consistency and identify modelling errors

7.3 Comparisons

Dymola and Simulink have their own advantages and disadvantages. The choice is up to user's requirement. Dymola (ver.5) and Simulink (ver.6) are compared based on the authors' experience. They are listed as follows:

1. Platform

Both Dymola and Simulink can work on a platform such as Windows, UNIX and Linux. At least 150MB Disk space and 128MB RAM are needed for Dymola while 600MB Disk space and 256MB RAM are needed for MATLAB only.

2. Need for supporting software

Dymola provides self-contained environment for modelling and simulating. It doesn't need any other software to support it after installation. Simulink is built on the core MATLAB numeric, graphics, and programming functionality so it must run in the environment of MATLAB.

3. Modelling method

Dymola uses a modelling methodology based on object oriented modelling and equations. Users do not need to convert equations to block diagram. Simulink is based on many basic blocks. Users should convert equations to block diagram form manually.

4. Program language

Dymola uses Modelica language which is for modelling of large, complex and heterogeneous physical systems. Simulink uses M-file, which is based on MATLAB language and C-code for programming. It takes time to build complex physical systems.

5. Inheritable components and need for rewriting

Dymola uses Modelica language designed to build basic components. All the components are inheritable. Users can take advantage from available basic and other existing components, while new components are built up. Blocks of Simulink are not inheritable. If users want to use some features of certain block they have to rewrite the block.

6. Expandability and openness

Both Dymola and Simulink have an open architecture. It is possible to expand components and blocks infinitely.

7. Interface

Both Dymola and Simulink support icon dragging. Users connect different components or blocks using an interconnection line according to object physical structure or mathematic models. User doesn't need to write thousands of lines of codes.

8. Icons of components or blocks

In Dymola the icons of model components are defined either by drawing shapes, or by importing graphics from other tools in bitmap format could be both either 3-D and 2-D style. In Simulink users can only write commands to draw 2-D icons.

9. Model editor

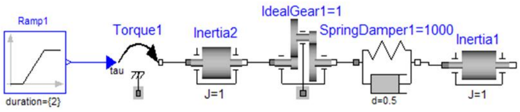
In Dymola modelling editor components cannot be dragged among different windows while in Simulink they can be. There is a pop-up menu for Simulink to cut, delete, copy and paste components while there is no such menu in Dymola.

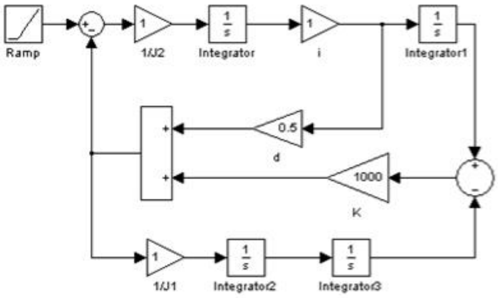
10. Algebraic loops

In Simulink the basic equations can not be simulated directly due to algebraic loop, which only contains algebraic equations. It is necessary for users to transform the equations to the form required by Simulink. In Modelica it is possible to write balance and other equations in their natural form as a system of differential-algebraic equations.

11. Difference in model generation practices
Because Dymola is an object-oriented application the differences in model generation practice are smaller than in Simulink where there are more modelling steps and thus more options to choose.
12. Visualization of simulation results
There are two ways in Dymola to visualize simulation results: plotting and animation. Only the finished calculation results can be displayed in Dymola. In Simulink it is easy to observe data processing either during or after finishing calculations through displays and scopes of Virtual Reality Toolbox. This facility can help users to gain deeper insight into complex 3-D motion of dynamic system but it is available only as an separate option.
13. Result analysing capabilities
MATLAB is a powerful tool to calculate, plot and analyse data. Simulink results can be easily edited and analyzed through MATLAB commands or GUI interface. Dymola enables user to visualize simulation results rather easily but it seems to be less powerful tool to analyze and calculate.
14. Code generation comparison
C-code can be generated from simulation models by both using Dymola and Simulink. Users can use this highly efficient code to write their own control program for specialized Hardware-in-the-Loop (HIL) systems, such as, dSPACE, RTLAB, xPC and others. Code can also be generated by Dymola for simulation in Simulink. Code from Simulink to Dymola needs Simelica tool.
15. New component development
If there are no components to satisfy the requirements of users, then new components should be developed. In Dymola equations have to be written with Modelica language while in Simulink user only needs to connect blocks together.
16. Price comparison
The price for one license of the Dymola Standard Version on a PC operating under Windows98/ NT/ 2000/ XP is EUR 1600 + VAT for universities [7]. The price for MATLAB/Simulink consists of price of many Toolboxes. Some Toolboxes are very expensive. The total price list can be obtained from [8]. Generally speaking MATLAB/ Simulink is more expensive than Dymola.

Fig.7.2 shows models built in Dymola and Simulink for the same object.





Dymola Model Simulink Model

Figure 7.2 The models for same object built in Dymola and Simulink

According to the comparison of the two softwares Dymola should be used for complex models with tens of thousands of equations mostly from multibody domain. It is easier to use than MATLAB/ Simulink due to its “object orientation”. However, Dymola is less powerful than MATLAB/ Simulink in calculation, data analysis and result editing.

7.4 Illustration of analytical modelling approaches

The following approaches are often used to solve behaviour of dynamical systems:

a) Analytical dynamics

Equations of motion are derived by Lagrange’s dynamics or some other theoretical method. Then they are solved by a numerical algorithm.

b) Dynamical equations are written in block structure form like in Matlab and then solved.

c) Object oriented method like in Dymola.

Now the analytical approach is illustrated. The simplified power transmission system is shown in Fig.7.3. It is similar to the model presented in Fig. 7.3.

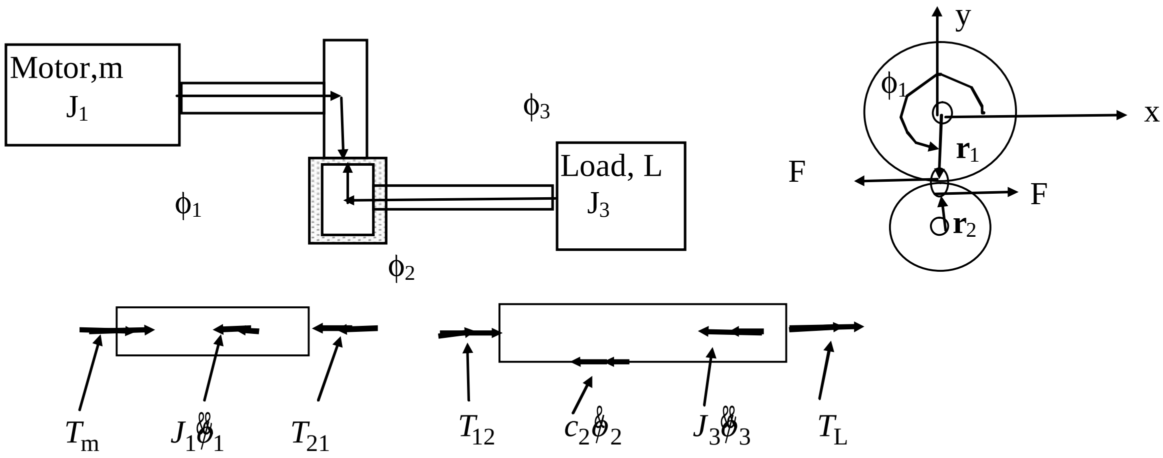


Figure 7.3 Analytical approach for modeling the dynamic equations of a simple gear transmission

Lagrange's function is defined as the difference between kinetic energy *T* and potential energy *V*. Other forces, which are not derivable from potential energies, are derived using the principle of virtual work.

$$
\begin{aligned}
&L = T - V \\
&\dot\phi_2 = n\dot\phi_1 \rightarrow \phi_2 = n\phi_1 \rightarrow \phi_2 = \frac{r_1}{r_2}\dot\phi_1 \\
&L = \tfrac{1}{2} J_1\dot\phi_1^{\,2} + \tfrac{1}{2} J_3\dot\phi_3^{\,2} - \tfrac{1}{2} k\left(\phi_3 - \phi_2\right)^2 \\
&D = \tfrac{1}{2} c_2\dot\phi_2^{\,2}
\end{aligned}
\qquad (7.1)
$$

Here:
n is transmission ratio of the gear
D is dissipation function describing viscous energy dissipation at dof 2.
Nonlinear equations of motion can be obtained using Lagrange's dynamics for each degree of freedom. For example for the dof 1

$$
\phi_1 \quad : \quad \frac{d}{dt}\left[\frac{\partial L}{\partial \dot\phi_1}\right] - \frac{\partial L}{\partial \phi_1} = Q_1 \quad , \quad Q_1 = \sum_k \vec{F}_\mathrm{k} \bullet \frac{\partial \vec{R}_\mathrm{k}}{\partial \phi_1} \qquad (7.2)
$$

Now the mass of the gearwheel 1 is neglected. External force on the dof 1 is tangential force *F* on the gear contact point. Its position vector in the cartesian xyz coordinates is

$$\mathbf{R} = z\hat{\mathbf{k}} + r_1 e^{i\phi_1} = \begin{bmatrix} r_1\cos\phi_1 \\ r_1\sin\phi_1 \\ z \end{bmatrix},$$

$$Q_1 = \begin{bmatrix} -F \\ 0 \\ 0 \end{bmatrix} \bullet \frac{d}{d\phi_1}\begin{bmatrix} r_1\cos\phi_1 \\ r_1\sin\phi_1 \\ z \end{bmatrix} = \begin{bmatrix} -F \\ 0 \\ 0 \end{bmatrix} \bullet \begin{bmatrix} -r_1\sin\phi_1 \\ r_1\cos\phi_1 \\ 0 \end{bmatrix} = Fr_1\sin\phi_1 \tag{7.3}$$

$$\phi_1 = 3\tfrac{\pi}{2}, \quad Q_1 = T_{21} = -Fr_1 \quad \Rightarrow \quad Q_2 = T_{12} = +Fr_2$$

$$n = \frac{r_1}{r_2} = \frac{T_{21}}{T_{12}} \rightarrow$$

Equation of motion for the dof 1 is

$$\phi_1 \quad : \quad J_1\ddot{\phi}_1 = T_\mathrm{m} - T_{21} = T_\mathrm{m} - Fr_1 = T_\mathrm{m} - nT_{12} \tag{7.4}$$

Equation of motion for dof 2 is

$$\begin{aligned} &\phi_2 \quad : \quad \frac{d}{dt}\left[\frac{\partial L}{\partial\dot{\phi}_2}\right] - \frac{\partial L}{\partial\phi_2} = Q_{2_\mathrm{i}} \quad , \quad Q_2 = \sum_k \vec{F}_\mathrm{k} \bullet \frac{\partial\vec{R}_\mathrm{k}}{\partial\phi_2} \\ &\phi_2 \quad : \quad J_2\ddot{\phi}_2 = T_{12} - c_2\dot{\phi}_2 - k(\phi_2 - \phi_3) = 0 \end{aligned} \tag{7.5}$$

Equation of motion for the dof 3 is

$$\begin{aligned} &\phi_3 \quad : \quad \frac{d}{dt}\left[\frac{\partial L}{\partial\dot{\phi}_3}\right] - \frac{\partial L}{\partial\phi_3} = Q_{3_\mathrm{i}} \quad , \quad Q_3 = \sum_k \vec{F}_\mathrm{k} \bullet \frac{\partial\vec{R}_\mathrm{k}}{\partial\phi_3} \\ &\phi_3 \quad : \quad J_3\ddot{\phi}_3 + k(\phi_3 - \phi_2) = T_\mathrm{L} \end{aligned} \tag{7.6}$$

Equations of motion for the system are

$$\begin{aligned} &J_1\ddot{\phi}_1 = T_\mathrm{m} - n\left(c_2 n\dot{\phi}_1 + k(n\phi_1 - \phi_3)\right) \\ &J_1\ddot{\phi}_1 + c_2 n^2\dot{\phi}_1 + k\left(n^2\phi_1 - n\phi_3\right) = T_\mathrm{m} \\ &J_3\ddot{\phi}_3 + k(\phi_3 - n\phi_1) = T_\mathrm{L} \\ &\begin{bmatrix} J_1 & 0 \\ 0 & J_3 \end{bmatrix}\begin{bmatrix} \ddot{\phi}_1 \\ \ddot{\phi}_3 \end{bmatrix} + \begin{bmatrix} n^2c_2 & 0 \\ 0 & 0 \end{bmatrix}\begin{bmatrix} \dot{\phi}_1 \\ \dot{\phi}_3 \end{bmatrix} + \begin{bmatrix} kn^2 & -kn \\ -kn & k \end{bmatrix}\begin{bmatrix} \phi_1 \\ \phi_3 \end{bmatrix} = \begin{bmatrix} T_\mathrm{m} \\ T_\mathrm{L} \end{bmatrix} \end{aligned} \tag{7.7}$$

It can be seen that the mass matrix is diagonal. Thus this system can be easily solved by solving first the highest derivatives.

8 DISCUSSION

One of the aims of this report was to find the way how two transmissions systems function and compare them by using simulation program Dymola. Both these transmission systems are based either directly or indirectly on patented inventions. Their descriptions were in many places vague and ambivalent. One very important and challenging preliminary task was to derive the kinematical sketches from rather unclear patent drawings and verbal descriptions. This revealed a clear need for more modern education for inventors and patent analysts.

The present simulations revealed that some transmission systems, which are presented in patents do not work as described in the patents or preliminary patent. One reason may be the vague description and the other reason may be that the system does not function at all. Valid and clear sketch maps are a requirement for accurate and reliable simulations to check their functionality. This approach should be a standard tool in checking and rejecting not working inventions and also developing further the chosen patented inventions into optimum direction.

A lot of useful results can be obtained from the sketch maps of the two transmissions systems presented in this research. First, transmission ratios are derived from speed equations assuming that torque is balanceable. In future work dynamic equations should be deduced for describing transient loadings at start-up and at braking. Second, to find the real behaviour of any proposed transmission both simulations and measurements of prototypes should be made. In one case the simulation model gave results, which are different from that claims of the inventor. Third, some features of Dymola and Simulink are not explained well enough in the manuals. This is a drawback since it requires undue effort of the user. It seems profitable that comparisons between the two softwares should be continued from the point of view of the needs, which user desires and also the experiences of users should be fed back to program developers in closer cooperation to get products, which are highly satisfactory for the end user.

9 SUMMARY

In this report two competing models of all mechanical power transmission systems have been presented. Transmission ratios of the two transmission systems are calculated in condition that torque is balanceable. Simulation models are built into the Dymola simulation environment. Different work condition and connection of transmission are assumed. Based on simulation results original models are modified. Wide and continuous variable transmission ratio is available in modified model of transmission system B. This is very similar to author's description. Large and constant transmission ratio can be achieved with a small sizes gear through modified transmission system A. From the comparison of the two transmission systems, transmission system B can be used more widely than modified transmission system A. Two simulation tools, Dymola and MATLAB/ Simulink, were studied. Some features of the two softwares were compared. Dymola proved to be more cost-effective, easier to use but it has less functions than MATLAB/ Simulink. Future work is suggested. Continuous work will be carried out based on the results presented in this research report.

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