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A CAD System for the Optimization of Gear Ratios for Automotive Automatic Transmissions

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A CAD System for the Optimization of Gear Ratios for Automotive Automatic Transmissions

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Abstract

An interactive design system has been developed for the design of automatic automotive transmission gear trains that can provide at least three forward and one reverse speed ratios. This user-friendly windowing system can access help files, display the functional representation of a mechanism, optimize the gear ratios and present the numerical results. The optimization procedure to find the optimum gear ratios and the corresponding number of gear teeth uses the Augmented Lagrangian Multiplier Method and can be applied to all epicyclic gear trains having two sets of three gears in which a ring gear is connected to a sun gear through a planetary gear. The Simpson and General Motors THM 440 gear trains are used to demonstrate the methodology. The gear teeth combinations are found such that they achieve the optimized gear ratios to within $\pm 1\%$ and satisfy the geometric constraints. PHIGS graphics functions are used in program routines to display the functional schematic of an epicyclic gear train on the computer screen. These routines are written in such a manner that other types of gear combinations can be displayed by simply adding additional modules to represent these new gear elements.

1 Introduction

Examining existing transmissions Tsai, et al. [1] observed that only coaxial links of an epicyclic gear train (EGT) are used as the input, output, or clutched to the frame to obtain the desired speed ratio[1]. To provide a minimum of three forward speed ratios and one reverse, it was found that EGTs should have four or more coaxial links. Tsai, et al. [1] have shown that there are only seven such non-isomorphic displacement graphs from which all EGTs with four or more coaxial links and up to six links can be derived. These can be grouped as one five-link chain with four coaxial links as shown in Figure 1(a) and six six-link chains with four or five coaxial links as shown in Figures 1(b)-1(g). Furthermore, it was shown that all practical six-link automatic automotive transmissions belong to the six non-isomorphic displacement graphs. The term automotive automatic transmissions has been used interchangeably with EGTs because we are dealing with all EGTs that can be potentially used as automotive automatic transmissions.

This paper concentrates on the following issues related to automotive automatic transmissions:

[1]Speed ratio is the ratio of the rotational speeds of the input link to the output link.

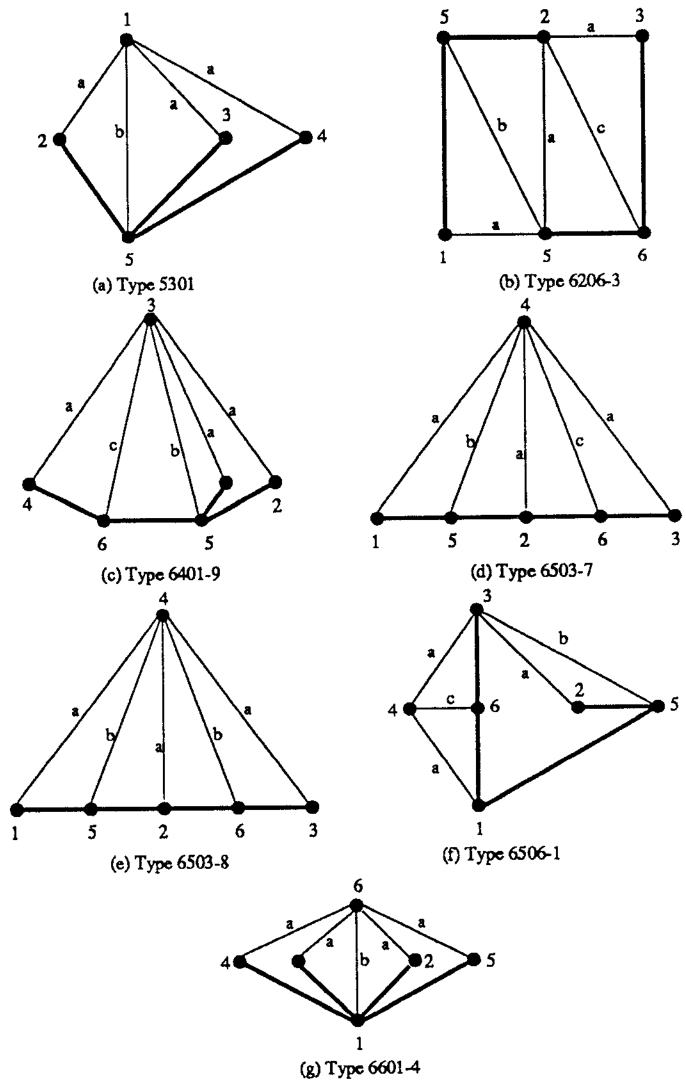


Figure 1: Graphs of gear trains with six or fewer links and with four coaxial links.

- Application of optimization techniques to find the gear ratios in automotive gear trains for a given set of reduction ratios.
- Development of a method of displaying epicyclic gear trains of different configurations.
- Development of a user-friendly, window-based interface for accessing the tools developed in the previous two stages.

The clutches, final reduction gears, and the differentials are not included in this study.

2 Optimization of gear ratios for EGTs

To achieve a specific set of speed ratios the designer has to choose a gear train with specific internal-external gear pair combinations, a set of clutches that are to be operated in a chosen sequence, and a set of gear ratios. The reduction ratio(speed ratio) is the ratio of the speed of the output link to the input link of a transmission mechanism. (In the automotive industry the reciprocal of the reduction ratio is often specified). The term gear ratio refers to the ratio of the numbers of teeth on two mating gears. The classical approach of finding the proper gear ratios has been to choose a gear train and a corresponding clutching sequence, and then to vary the gear ratios by trial and error until the best possible reduction ratios have been obtained. At the end of this trial and error process the designer will not know whether the best possible reduction ratios with respect to the original requirements have been achieved. In practice, however, one wants to choose a set of gear ratios to achieve a set of reduction ratios.

2.1 Formulation of Optimization Problem

The optimization methodology for determining these gear ratios for a particular set of gear trains is now described. The methodology is then demonstrated for the Simpson gear train (Figure 24 in ref [1]) and the General Motors THM 440 gear train (Figure 2 in [1]), each with a different number of reduction ratios. The clutching sequence, which includes the brake clutches, is assumed to be the same as those mentioned in [1]. The choice of the gear pairs (internal-external, external-external, external-internal) is also assumed to be the same. The identification numbers for the graphs corresponding to the gear trains have also been retained.

In the gear trains cited above there are two sets of three gears each. Each set has a sun gear connected to a ring gear through a planetary gear. The optimization formulation developed can be applied to all the variations of EGTs that can be obtained by changing the clutching sequences, clutch location and the number of clutches. For example, the optimization problem can be equally applied to the gear train in the Hydramatic THM 440-T4 (Figure 2 in [1]), the Ford Axod (Figure 3 in [1]) and the Hydramatic 700-R4, which differ only in their clutch locations or number of clutches.

The optimization problem can also be set up for all the gear trains that fall into similar categories, which result from a systematic enumeration of EGT mechanisms resulting from the seven graphs using various combinations of internal-external, external-internal, and external-external gear meshes for each gear pair. The phrase similar category means the gear trains in which there are sets of three gears, each of which has a sun gear connected to a ring gear through a planetary gear.

Graph will be used to represent the functional schematic of a mechanism. In a graph representation, vertices denote links and edges denote pair connections between links. In order to distinguish different pair connections, heavy edges are used to denote gear pairs and thin edges are used to denote turning pairs. Let i, j denote two adjacent vertices of a geared edge in a graph [2] in which k is the associated transfer vertex. Let ω_i, ω_j, and ω_k denote the angular velocities of links i, j, and k, respectively. Let r_{ji} denote the gear ratio of the gear pair on links j and i, either positive or negative depending on whether the mesh is internal or external; that is, $| r_{ji} | = t_j/t_i$ where t_j and t_i denote the number of teeth of the gear pair on links j and i, respectively. Then links i, j, and k constitute a simple epicyclic gear train for which k is the gear carrier. The angular velocities of these elements are related by the linear equation

$$\omega_i - r_{ji}\omega_j + (r_{ji} - 1)\omega_k = 0 \tag{1}$$

For a one degree-of-freedom EGT with n links, there are $(n - 2)$ geared edges. Equation (1) is written $(n - 2)$ times, one for each of the basic EGTs. Together, these equations comprise the rotational displacement equations of the epicyclic chain.

For a six-link, one degree-of-freedom EGT, the rotational displacement equations represent a set of four simultaneous equations with six variables $\omega_1 \ldots \omega_6$. If two of these variables are specified, and if the gear ratios are given, then we can solve for the remaining variables. That is, the rotational speeds of the remaining elements in the gear train are uniquely specified, including the output link. Note that the set of simultaneous equations to be solved depends on the two input variables, which in turn depend on the clutch conditions.

Assuming that the element clutched to the input link has a rotational speed of 1 rpm, and the elements for which a brake has been applied has a rotational speed of 0 rpm, then the rotational speed of the output element gives the reduction achieved by the gear train with the given clutch selection (or the reduction ratio). Usually a reduction ratio is less than 1, but if the gear train functions as an overdrive it can be greater than 1.

We define an objective function F in terms of the reduction ratios, R_i, $i = 1,2 \ldots n$ for the gear train, the minimization of which ensures that the best possible gear ratios are achieved, subject to certain constraints. Thus, we minimize

$$F(\mathbf{R}) \tag{2}$$

subject to:

$$g_j(\mathbf{R}) \leq 0 \quad ; \quad j = 1, m \tag{3}$$

$$h_k(\mathbf{R}) = 0 \quad ; \quad k = 1, n \tag{4}$$

where g_j represent inequality constraints and h_k represent equality constraints. The objective function is defined as

$$F(\mathbf{R}) = \sum_{i=1}^{n} (R_i - k_i)^2$$

where R_i is the ith reduction ratio parameter obtained by the minimization process, k_i is the ith desired reduction ratio, and n is the number of reduction ratios. The reduction ratios $R_1 \ldots R_n$ depend on the clutch selection and the topology of the gear train. The constraints under which the function $F(R)$ is minimized will depend on the gear train being considered.

2.1.1 Formulation for the Simpson Gear Train

Consider the stick diagram for the Simpson gear train shown in the Figure 2(a). The graph for the gear train is shown in Figure 2(b). The clutching sequence is shown in Figure 2(c), in which the letter C_i represents a rotating clutch and B_i represents a brake. The rotational displacement equations for this graph can be written as [3, 4]

$$[A][W] = [0] \tag{5}$$

where

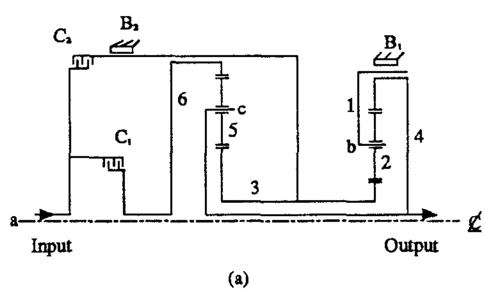
$$A = \begin{bmatrix} r_{42}-1 & 1 & 0 & -r_{42} & 0 & 0 \\ r_{32}-1 & 1 & -r_{32} & 0 & 0 & 0 \\ 0 & 0 & -r_{35} & r_{35}-1 & 1 & 0 \\ 0 & 0 & 0 & r_{65}-1 & 1 & -r_{65} \end{bmatrix}$$

and

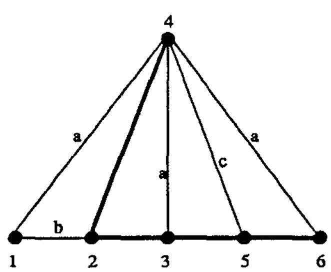
$$W = \begin{bmatrix} \omega_1, & \omega_2, & \omega_3, & \omega_4, & \omega_5, & \omega_6 \end{bmatrix}^T$$

Solving the system of Eqs. (5) four times, each time for a different clutching sequence using the table shown in Figure 2(c), and expressing the reduction ratios in terms of the gear ratios it can be shown that the reduction ratios are given as

$$R_1 = r_{65}r_{32}/(r_{65}r_{32} + (r_{42} - r_{32})r_{35}) \tag{6}$$
$$R_2 = r_{65}/(r_{65} - r_{35}) \tag{7}$$
$$R_3 = 1 \tag{8}$$
$$R_4 = r_{32}/r_{42} \tag{9}$$



(a)



(b)

Gear	C1	C2	B1	B2
FIRST	x		x	
SECOND	x			x
THIRD	x	x		
REVERSE		x	x	

C - Input Clutch, B - Brake

(c)

Figure 2: Simpson gear train, (a) functional schematic, (b) graph 6206_3, (c) clutching sequence.

It is noted that in an automotive automatic transmission the output link shown in Figure 2(a) is connected to a final reduction gear set and then to the differential gear. Under certain clutch combinations the gear train may act as a rigid body, in which case the reduction ratio will be equal to one, and the designer will not be able to affect it by his selection of gear ratios. This is known as the direct drive and is chosen purposely to maximize the transmission efficiency. It should be realized that for an arbitrary combination of k_i there may not be a set of solutions for the gear ratios r_{ji} that satisfy Eqs. (6) to (9). In other words, the ideal minimum value of the cost function ($F(\mathbf{R}) \rightarrow 0$) may or may not be achievable. The constraints under which the function $F(\mathbf{R})$ is minimized for the Simpson gear train are now described. Let r_i, d_i, and t_i be the radius, diameter and number of teeth of gear element i. If the diametral pitch P of all the gears is the same, then $t_i = d_i/P$ and we have for the Simpson gear train $r_6 = r_3 + 2r_5$ or

$$r_{65} + r_{35} = 2 \tag{10}$$

and in a similar manner

$$r_{42} + r_{32} = 2 \tag{11}$$

Note that both r_{35} and r_{32} are negative numbers, since they correspond to external gear meshes i.e. $r_{35} = -r_3/r_5$ and $r_{32} = -r_3/r_2$.

A practical gear train cannot have too large or too small gear ratios. For the Simpson gear train we require the gear ratios to fall within the following limits :

$$\begin{array}{rcccr} 3.0 & \leq & r_{65} & \leq & 8.0 \\ -6.0 & \leq & r_{35} & \leq & -1.0 \\ 3.0 & \leq & r_{42} & \leq & 8.0 \\ -6.0 & \leq & r_{32} & \leq & -1.0 \end{array}$$

The sun gear is usually larger than the planet gear, and it accommodates the center shaft. Also, to avoid dynamic imbalance and to facilitate load sharing there are usually more than one planetary gears acting as parallel conduits of power transfer. Planet gears can thus be smaller; hence the limits on r_{65} and r_{35}. These limits can be changed if desired.

The optimization problem now is to minimize

$$F(\mathbf{R}) = \sum_{i=1}^{4} (R_i/k_i - 1)^2 \tag{12}$$

subject to the constraints

$$g_1 : \quad r_{65}/8.0 - 1 \tag{13}$$

$$g_2 : \quad 3.0/r_{65} - 1 \tag{14}$$

$$g_3 : \quad (-r_{35})/6.0 - 1 \tag{15}$$

$$g_4 : \quad (1.0)/(-r_{35}) - 1 \tag{16}$$

$$g_5 : \quad r_{42}/8.0 - 1 \tag{17}$$

$$g_6 : \quad 3.0/r_{42} - 1 \tag{18}$$

$$g_7 : \quad (-r_{32})/6.0 - 1 \tag{19}$$

$$g_8 : \quad 1.0/(-r_{32}) - 1 \tag{20}$$

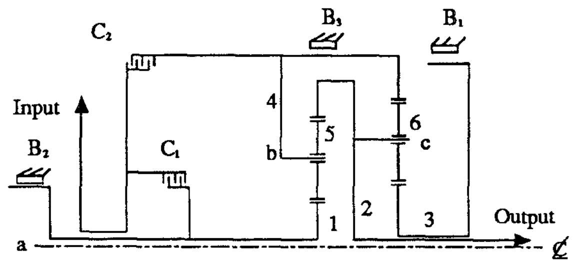
$$h_1 : \quad (r_{65} + r_{35})/2 - 1 \tag{21}$$

$$h_2 : \quad (r_{42} + r_{32})/2 - 1 \tag{22}$$

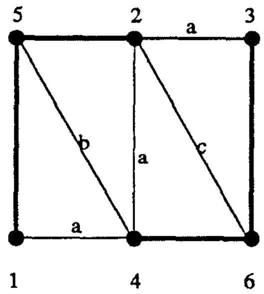
The quantities R_1, R_2, R_3, and R_4 are given in Eqs. (6)–(9). Note that all quantities have been scaled to have comparable magnitude before they are used, and that the inequality constraints with upper and lower limits have each been split into two single-limit constraints.

2.1.2 Formulation for a THM 440 Gear Train

The stick diagram for the General Motor's THM-440 gear train is shown in Figure 3(a). The graph for the gear train is shown in Figure 3(b) and the clutching sequence for the gear train is shown in the Figure 3(c). The optimization problem for this gear train is to minimize



(a)



(b)

	C_1	C_2	B_1	B_2	B_3
FIRST	X		X		
SECOND		X	X		
THIRD	X	X			
FOURTH		X		X	
REVERSE	X				X

C - Input Clutch, B - Brake

(c)

Figure 3: Hydra-Matic THM 440-T4 (a) functional schematic (b) graph 6206_3 (c) clutching sequence.

$$F(\mathbf{R}) = \sum_{i=1}^{5} (R_i/k_i - 1)^2 \tag{23}$$

where

$$R_1 = r_{15}r_{46}/(r_{15}r_{46} + (r_{25} - r_{15})r_{36}) \tag{24}$$

$$R_2 = r_{46}/(r_{46} - r_{36}) \tag{25}$$

$$R_3 = 1 \tag{26}$$

$$R_4 = 1 - r_{15}/r_{25} \tag{27}$$

$$R_5 = r_{15}/r_{25} \tag{28}$$

Table 1: Table of optimized gear ratios for sample inputs for the Simpson gear train.

Set #	Desired Overall Reduction Ratio		Achieved Overall Reduction Ratio		F(**R**)	Optimized Gear Ratios	
1	k_1	0.3521	R_1	0.352189	1.68×10^{-7}	r_{65}	4.996
	k_2	0.6250	R_2	0.625115		r_{35}	2.996
	k_3	1.0000	R_3	1.000000		r_{42}	3.874
	k_4	-0.4839	R_4	-0.483897		r_{32}	1.874
2	k_1	0.3000	R_1	0.326227	4.92×10^{-2}	r_{65}	6.011
	k_2	0.7500	R_2	0.599792		r_{35}	4.011
	k_3	1.0000	R_3	1.000000		r_{42}	3.826
	k_4	-0.5000	R_4	-0.477251		r_{32}	1.826
3	k_1	0.4000	R_1	0.400004	3.76×10^{-8}	r_{65}	3.499
	k_2	0.7000	R_2	0.700006		r_{35}	1.499
	k_3	1.0000	R_3	1.000000		r_{42}	3.333
	k_4	-0.4000	R_4	-0.399994		r_{32}	1.333

Replacing r_{65}, r_{35}, r_{42} and r_{32} by r_{25}, r_{15}, r_{46} and r_{36}, respectively, in the constraint Eqs. (13)–(22) gives the constraints equations for this problem. The two optimization problems were solved by the Augmented Lagrangian Multiplier Method [5] together with the Conjugate Gradient Method [6] and Golden Section Search Method [6].

2.2 Results and Discussion

The optimization program was written in C language. The designer initially specifies the gear train type and stipulates the reduction ratios to be achieved. The optimization program starts at an arbitrary design point and at the end of its execution displays the optimized gear ratios. Results for sample inputs for the Simpson and THM-440 gear trains are shown in Tables 1 and 2, respectively.

To verify the procedure, the results of Tsai [7] were used. Tsai has developed an algorithm for the automatic generation of the rotational displacement equations, if the adjacency matrices are given, and has demonstrated his algorithm for the angular velocity analysis of an EGT with any number of links. He started with a set of gear ratios for the Simpson gear train and arrived at the reduction ratios and the angular velocity of other links in the gear train. Set # 1 of the Simpson gear train given in the Table 1 starts with the reduction ratios obtained by Tsai. The optimization program obtained the same gear ratios that was the input for Tsai's procedure. These values are shown in the last column of Table 1.

The design is far from complete at this stage. Once the gear ratios are obtained, further steps in the design's analysis are needed to determine the gear thickness, material, teeth number, the shaft diameters, etc., based on the power transmitted.

To relate the gear ratios to the gear teeth numbers, a program was written to generate a list of gear ratios for all combinations of gear teeth from 14 to 120. These ratios have been sorted and are used to determine the sets of gear teeth numbers that would give gear ratios to within $\pm1\%$ of the optimized gear ratios. (The value of 1% was based on a sensitivity analysis that found the effect

Table 2: Table of optimized gear ratios for sample inputs for the THM 440 gear train.

Set #	Desired Overall Reduction Ratio		Achieved Overall Reduction Ratio		F(**R**)	Optimized Gear Ratios	
	k_1	0.3425	R_1	0.343011		r_{25}	3.498
1	k_2	0.6269	R_2	0.635171	1.00×10^{-5}	r_{15}	1.498
	k_3	1.0000	R_3	1.000000		r_{46}	4.699
	k_4	1.4286	R_4	1.428328		r_{36}	2.699
	k_5	-0.4286	R_5	-0.428328			
	k_1	0.4000	R_1	0.400547		r_{25}	3.369
2	k_2	0.7000	R_2	0.698090	4.16×10^{-3}	r_{15}	1.369
	k_3	1.0000	R_3	1.000000		r_{46}	3.524
	k_4	1.5000	R_4	1.406424		r_{36}	1.524
	k_5	-0.5000	R_5	-0.406424			

Table 3: Gear teeth values for obtaining optimized ratios r_{65}, r_{35}, r_{42}, and r_{32} within $\pm1\%$ for Set # 1 in Table 1.

t_6	t_5	r_{65}	t_3	t_5	r_{35}	t_4	t_2	r_{42}	t_3	t_2	r_{32}
70	14	5.000	42	14	-3.000	66	17	3.882	32	17	-1.882
80	16	5.000	48	16	-3.000	62	16	3.875	30	16	-1.875
100	20	5.000	60	20	-3.000	58	15	3.867	28	15	-1.867
120	24	5.000	72	24	-3.000						
⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮	⋮

on the reduction ratios due to a perturbation of the gear ratios for the Simpson and the THM 440 gear trains considered).

In the Simpson gear train, r_{42} and r_{32} have element 2 as a common gear, and r_{35} and r_{65} have element 5 as a common gear. Similarly, for THM 440 gear train r_{25} and r_{15} have element 5 as a common gear and r_{46} and r_{36} have element 6 as a common gear. Also, the sum of the number of teeth of the sun gear and twice the number of teeth on the planet gear should equal the number of teeth on the ring gear. This is equivalent to all the constraints of the type specified by Eqs. (21) and (22). The sets of gear teeth now are narrowed to the set that can accommodate the same number of gear teeth for the common gear and can satisfy the geometry constraint given by Eqs. (21) and (22). Carrying out this search to obtain the first set of optimized gear ratios in Table 1 for the Simpson gear train yields the values shown in Table 3. Tsai [7] started with the gear teeth number set $t_6 = 70$, $t_5 = 14$, $t_3 = 42$, $t_4 = 62$, $t_2 = 16$, and $t_3' = 30$. (Element 3 for the simpson gear train is a multiple gear). Referring to Table 3 it can be seen that other combinations are possible to give the reduction ratios to be exactly equal or very close to the values specified by set # 1 in Table 1.

The approach used to find the teeth number of the gears must be weighed against the changes that may have to be made during the gear analysis based on transmitted power. Reduction ratios that can be achieved may have to be modified with respect to the optimized values of the gear and reduction ratios. Hence, an iterative procedure between the values given in Table 3 and those required to satisfy the transmitted power may have to be developed in order for the process to be fully automated.

3 Functional Representation of EGTs

This section deals with the methodology that has been developed for displaying an EGT. Both PHIGS[2] and C have been used to obtain the display of various elements comprising the EGT. PHIGS allows for the creation, storage and dynamic modification of these images [8], is device-independent and is based on the concept of an abstract workstation.

Currently, the representation of gear trains has been limited to 2D cross sectional views of the gear trains. However, the PHIGS function calls are made in 3D to facilitate upgrading at a later stage. Each EGT is displayed along with its corresponding graph.

Consider a PHIGS square display window with the length of its side equal to one. Any image that is displayed will be correspondingly scaled to fit in this window. Once the graph number is known, the graph is displayed in the upper left corner of the display window, and then the functional schematic is scaled to fit in the remaining area. The turning pair adjacency matrix, gear pair adjacency matrix, and axes levels obtained for a given graph are assumed known.

3.1 Element Modules

An EGT is considered to be made up of several elements (internal gear, external gear, etc.). To draw the functional representation of the EGT information concerning the number of elements, type of elements and the data for each element must be known. Each element is identified by a number. At present, four major modules have been written to create PHIGS structures for the following element types: internal gear, external gear, carrier and shaft. More modules for different element types can be easily added, and each module itself can be upgraded to represent an element with more details and complexity. The four element type modules are described below, with the internal gear module described in more detail. Based on the specific gear train to be displayed, the appropriate data files are read by the program to obtain the element's parameters, their orientation, etc., in a specific order. Each element is described with respect to its local coordinates and since the relative location of each element is known with respect to the origin of a complete gear train the appropriate transformations are performed to generate the display.

3.1.1 Internal Gear Module

The internal gear is composed of the parameters shown in Figure 4(a) . The coordinate locations of the vertices that are numbered are described in terms of the parameters shown with respect to the local origin. Line 1–10 may or may not be drawn, depending on whether or not the shaft has a fixed or rotating center. The options that can be specified for drawing an internal gear are: (i) the orientation of the gear, the gear can be facing left or right; (ii) the presence or absence of a shaft

[2]PHIGS is an acronym for Programmer's Hierarchical Interactive Graphics System. PHIGS standard is an international graphics programming standard developed by the International Standards Organization (ISO) and the American National Standards Institute (ANSI).

at its center; and (iii) when the shaft is present, whether it is inside or outside the internal gear (such a shaft usually acts as a carrier to some other element in the gear train.) These options are shown in Figure 4(b). If a hub is not present 'hub length' is specified as half the arm thickness.

3.1.2 Multiple Gear Module

Only a multiple gear with external gears is considered. The multiple gear is shown in Figure 5. The coordinate locations of the vertices that are numbered are described in terms of the parameters shown with respect to the local origin. The multiple gear can be either a double or triple gear. All gears have a turning pair at their centers.

3.1.3 External Gear Module

The external gear is shown in Figure 6. The coordinate locations of the vertices that are numbered are described in terms of the parameters shown with respect to the local origin. Option 1 selects either a fixed or rotating center for gear, and option 2 indicates whether the hub is to the right or left of the local origin.

3.1.4 Carrier

The carrier is shown in Figure 7. The coordinate locations of the vertices that are numbered are described in terms of the parameters shown with respect to the origin. Option 1 selects the orientation of shaft number two, which can be either left or right with respect to the local origin. Rotating type joints are assumed at both ends of the carrier.

3.1.5 Shaft

A shaft is described by its diameter. A shaft can, however, be connected rigidly to any number of elements in the gear train. A general schematic of a shaft is shown in Figure 8(a). As shown in the figure the local origin is placed at the intersection of the center line of the shaft and the line passing through the middle of the width of the connection that is left-most on the shaft. The ends of a shaft are approximated as shown in Figure 8(b). By specifying the end type, the left and right ends of a shaft may or may not be drawn. The connections are assumed to be one of the three types shown in Figure 8(c).

3.2 Results and Discussion

Once all the relevant data are known, the program creates a structure network that is composed of several structures for the various elements in the gear train, with the specified transformations based on their relative locations. (The structure is only created, but not yet displayed.) The size of the gear train varies dynamically based on each set's dimensional data describing an EGT. The gear train structure is appropriately scaled and then displayed onto the screen.

To demonstrate the procedure a data set has been created for the Simpson gear train (Figure 24 in [1]) and the THM 440 gear train (Figure 2 in [1]). The results are shown in Figures 9 and 10, respectively.

Some researchers [9] are currently generating functional representations, including 3-D shaded images, of various types of mechanisms based on the graph representation. Such images are,

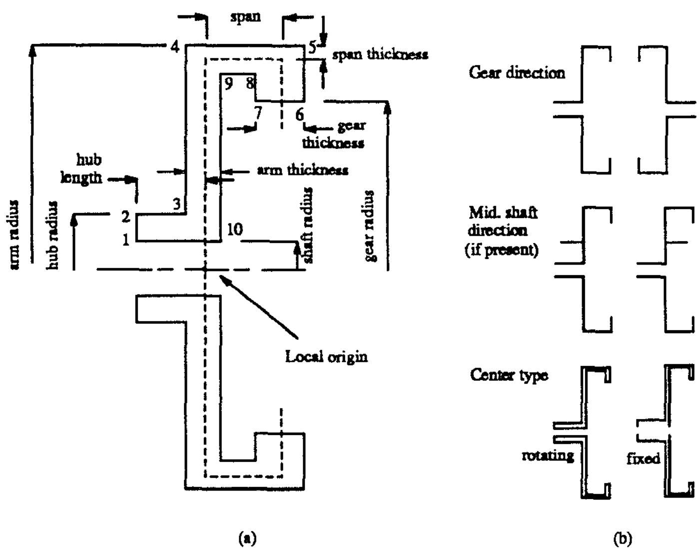


Figure 4: Internal gear (a) parameters (b) options.

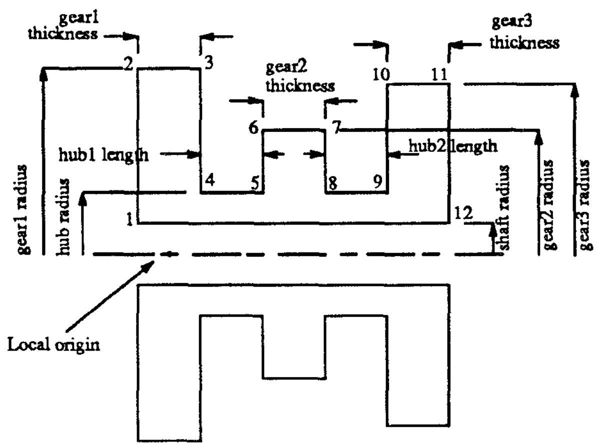


Figure 5: Multiple gear (a) parameters.

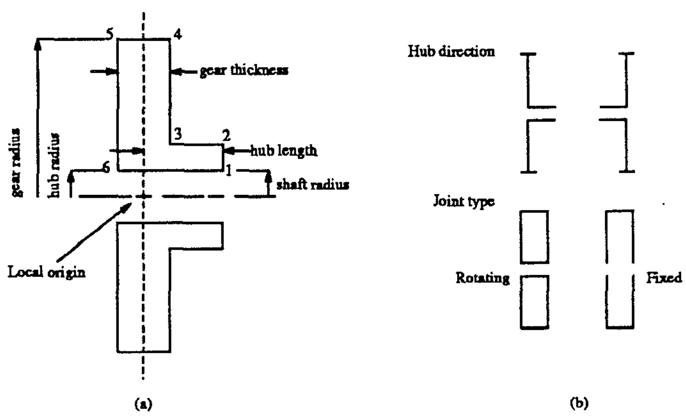


Figure 6: External gear (a) parameters (b) options.

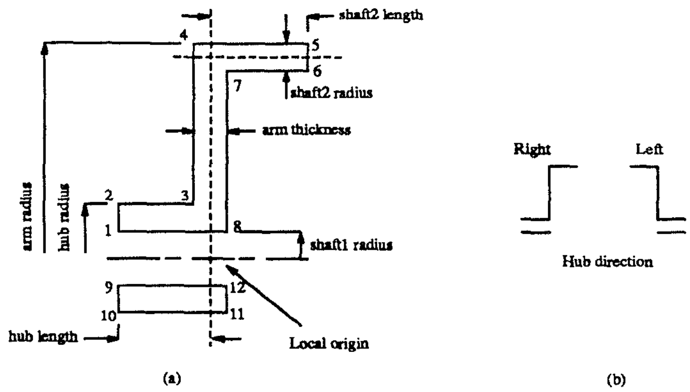


Figure 7: Carrier (a) parameters (b) options.

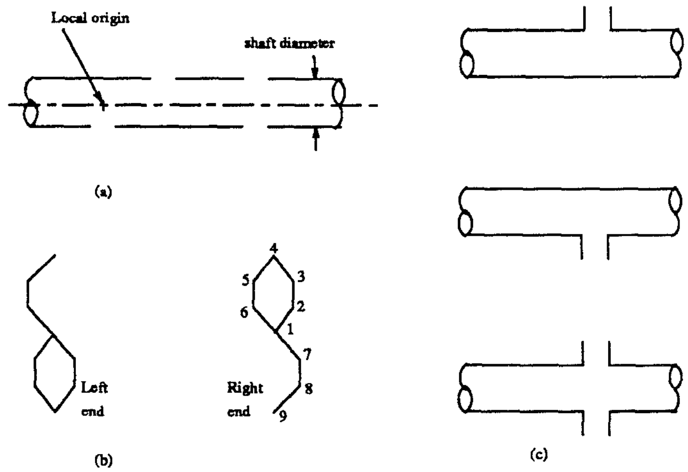


Figure 8: Shaft (a) typical cross-section (b) end approximation (c) connections.

however, limited to external-external gear pair combinations, although equivalent mechanisms can be obtained using external-internal gear pair combinations. Although the approach here does not generate 3-D shaded images of the gear trains, it is not limited to gear trains containing only external gears and it gives a cross section of the gear train that can be understood by any designer.

4 User Interface

A User Interface (UI) was developed to integrate the various tools previously discussed. That is, it provides access to the optimization and functional representation display routines written for this EGT design environment, and provides access to help files and present the numerical results.

5 Conclusions

We have concentrated on three issues related to automotive epicyclic gear trains.

First, a procedure for the functional and graph representation of EGT mechanisms on a computer screen using PHIGS has been developed and demonstrated for two existing automotive EGTs.

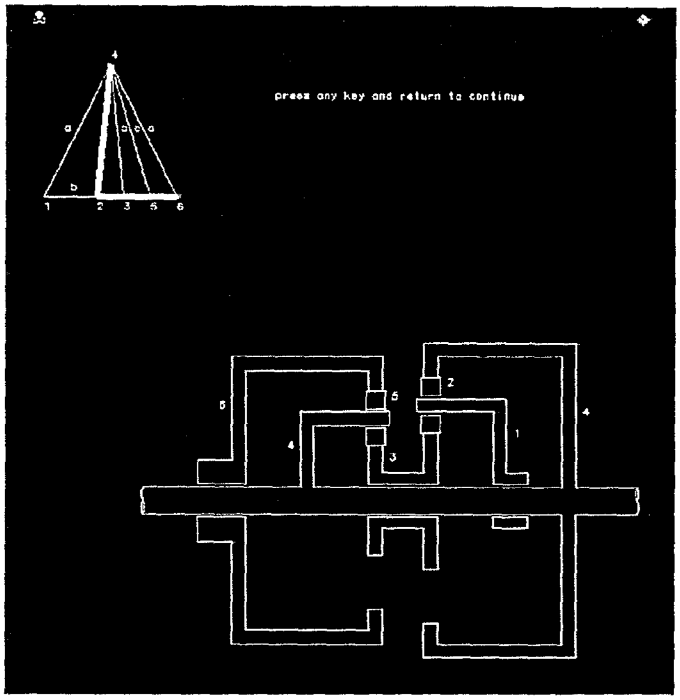


Figure 9: Functional representation of a Simpson gear train

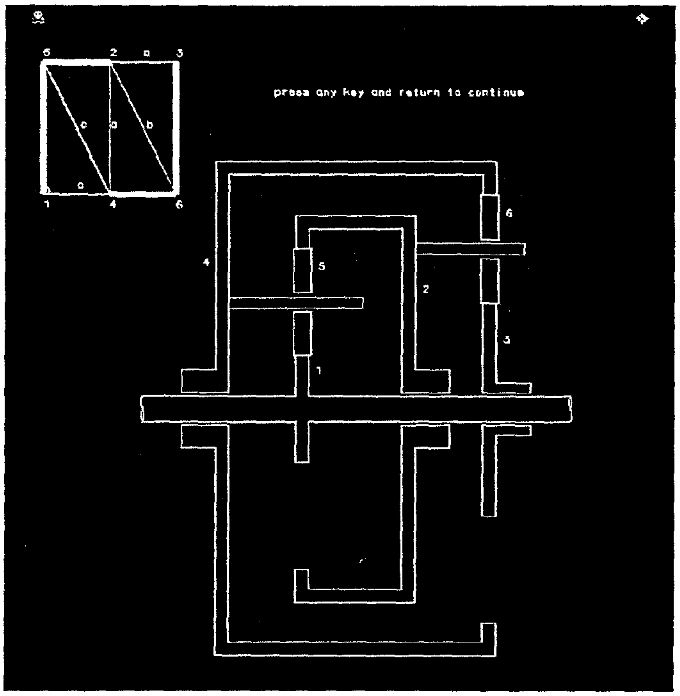


Figure 10: Functional representation of a THM 440 gear train

A limited number of element modules (external gear, internal gear, multiple gear, carrier, and shaft) have been developed to display the EGTs. Virtually all mechanisms that are comprised of these elements in various combinations can be displayed using these modules. To broaden its application to other mechanisms comprising different elements, additional modules can be incorporated. There is no limitation to the nature or complexity of the element that can be displayed.

Second, the traditional trial and error approach of finding the gear ratios for an automotive gear train to obtain a set of reduction ratios has been eliminated by formulating the task as a constrained nonlinear optimization problem. The optimization procedure was successfully applied to two automotive gear trains: the Simpson gear train and the THM-440 gear train. The optimization procedure finds those gear ratios for which the reduction ratios are closest to the original specification. Furthermore, the corresponding combination of gear teeth numbers that can satisfy these gear ratios to within 1% are also displayed. These gear teeth numbers also satisfy the geometric constraints.

Third, a user friendly user windows environment was developed to access the optimization results and to display the EGT.

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