Towards Automated Synthesis of Automatic Automated Transmission Designs

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ABSTRACT: The design of automatic transmissions is a challenging task due to complex power flows and high variability of architectures. To overcome those difficulties a methodology is proposed which uses computational design synthesis to generate transmission designs. The methodology is efficiently dealing with the complexity and variability of transmissions while the engineer is focusing on the application of his engineering knowledge. An analysis of 1022 patented transmissions is shown and highlights the trend of increasing complexity in order to achieve higher capability. The generation of innovative transmission designs is presented producing designs with improved properties.

KEY WORDS: Power transmission, architecture generation, design , modeling, optimization [A2]

1. INTRODUCTION

The design process of an automatic transmission is driven by its purpose. The transmissions main purpose is the optimal propagation of the engine power to the driven axis. In the case of an automotive application a transmission is required to have multiple forward and at least one reverse gear. In practice the design of new automotive transmission designs is often based on centuries of experience and existing designs⁽¹⁾.

Transmissions are designed with multiple requirements in mind. The top requirement is a long service life which in case of automatic transmissions is affected by the design of the planetary gears. Further three of the top five requirements are low production costs, an optimal range of ratios and optimal gear steps⁽¹⁾. However a low production cost can only be achieved with simple designs, while an optimal range of ratios and optimal gear steps require complex designs. Thus the transmission design process aims to find designs that maximize capability and minimize complexity.

Figure 1 shows the results of a patent analysis and highlights a trend in the design of automatic transmissions: In order to achieve higher capability a more complex transmission is designed. The graph shows patented transmission designs by their **capability** (i.e. their number of forward speeds) over their **complexity** (i.e. their number of clutches, brakes and gear contacts)¹. Moreover a technological boundary is shown, as the patented designs are only able to achieve a certain apparent maximum capability for every level of complexity.

Fig. 1 Patented automotive planetary transmission designs by capability over complexity. The size of the circles represents the amount of patented designs. The highlighted grey area at Complexity 8 is the focus of the presented work.

Presented in this work is a methodology able to efficiently generate transmission designs with the maximal capability using a fixed set of components. The presented methodology is applied to the generation of transmission design with complexity 8 (highlighted in Figure 1) with the goal to generate designs with four or more forward speeds and at least one reverse speed. The generation of three novel and one already patented four-speed transmission designs is presented. For each of the novel generated architectures the authors were not able to find patents that include the same designs.

¹ A component permanently attached to the casing is considered as a component connected to a permanently engaged brake.

The presented model-based design approach allows the designer to define the components of an automatic transmission. These components may hold information on functionality, connectivity, variability, design parameters and simulation models. From a library of such components a designer may choose to use two planetary gears (with their planetary gear ratios as design parameters), two clutches, and two brakes.

With these components defined, a constraint problem is set up and solved to generate transmission topologies. From these generated topologies invalid and unwanted solutions are filtered out by determining the number of available forward and reverse speeds and checking whether a topology can be manufactured. The planetary gear ratios which are defined as design parameters are optimized based on the results of a kinematic analysis of the power flows of the generated transmissions.

An efficient execution of these steps provides the designer with a set of automatically generated and optimal transmission architectures for an automotive application.

Section 2 provides a brief overview of related work on the subject. Section 3 describes the scientific methodology applied for this research. In Section 4 the model-based design approach is presented. Section 5 explains the topology generation process and presents the evaluation and optimization of the generated topologies in detail. In Section 6 the results of the application of the presented methodology are presented. Section 7 presents the current research towards automatically generated simulation models for drivecycle simulations. Finally Section 8 concludes the paper and describes possible future work.

2. RELATED WORK

Modeling transmission components or systems such that the resulting power flows can be analyzed is common practice. Examples of modeling practices are the lever analogy ⁽²⁾, bond graph modeling ^(3, 4) and canonical graph modeling ⁽⁵⁾. Other approaches define kinematic equations to analyze the power flows of a transmission system ^(6, 7).

Some research also attempts to generate transmission topologies. Königseder and Shea employ graph grammars to generate transmission topologies ⁽⁸⁾. This approach is however restricted by the topologic and parametric rules that drive the generation process. The used algorithm applies these rules iteratively and is not able to guarantee that all relevant solutions are generated in a finite set of executions.

An effort has been made to use canonical graph characteristics to enumerate all possible transmission topologies ⁽⁹⁾ and automatically analyze the overall transmission ratios of the generated topologies ⁽¹⁰⁾. However canonical graphs of

transmission systems do not represent clutches nor brakes. Therefore a new graph is required for each clutching condition.

Bayrak et al. present an approach to design hybrid transmission architectures which generates all possible lever analogy representations for a set of components and applies bond graph rules to determine viable solutions ⁽¹¹⁾. This exhaustive enumeration of possible solutions is not constrained by engineering knowledge and may create many solutions, which have to be deemed invalid by the application of bond graph rules. Based on these generated architectures Bayrak et al. present a genetic optimization approach in order to find optimal designs ⁽¹²⁾.

Raghavan et al. ⁽¹³⁾ present an approach which enumerates all possible kinematic combinations of transmission elements and then searches for viable candidates based on the transmission governing equations. This approach has proven to find novel transmission architectures, including the design for the 8L90 transmission produced by General Motors ⁽¹⁴⁾. Their generation requires the designer to decide how many shifting elements have to be engaged at any given time, constraining the design space unnecessarily.

A similar to the approach of is used by Zhang et al. for the generation of hybrid power-split transmissions with one or two planetary gears ^(15, 16). The process first exhaustively enumerates all possible kinematic combinations of elements and then filters out invalid or inferior designs through analysis and simulation. This methodology is also used by Zhuang et al. for the generation of power-split transmissions with three planetary gears ⁽¹⁷⁾. With three planetary gears there exist 1.67×10^{11} possible topologies. However in this application a subset of 19,085,220 topologies were enumerated initially and 14 designs were found through evaluation, that could exceed existing an existing design in terms of performance.

Moreover various research exists which aims to optimize topological and/or configuration parameters of gear trains and/or gear contacts ^(18, 19). Other research aims to optimize the efficiency of drivetrains with automatic planetary transmissions by optimizing the shifting behavior ^(20, 21).

The presented work does not alter shifting behavior or detailed gear contact parameters and instead shows that optimal designs for a transmission system can be achieved through the variation of the transmission topology and optimization of planetary ratios. Further this work presents an approach for the generation of transmission topologies based on design knowledge rather than an enumeration of all possibilities.

3. SCIENTIFIC METHODOLOGY

The aim of the presented methodology is the generation of novel, innovative transmission designs that provide a higher

capability without increasing the complexity of the system. For more complex designs more alternatives are possible. For this reason an initial application was chosen where architectures are composed from the same components as the simplest patented transmission designs.

Table 1 shows all patented designs with complexity 8 of the patent analysis of Figure 1. Again the complexity is defined here as the sum of shifting elements and gear contacts. It is obvious that the most of these simple designs use two single planetary gears, two clutches and two brakes. These designs provide up to four forward speeds and (as it is common in automobiles) exactly one reverse gear.

Table 1 Patented transmission designs with complexity 8.

PATENT	YEAR	PLANETARY GEARS	GEAR CONTACTS	CLUTCHES	BRAKES	FORWARD SPEEDS
US2865231	1953	2	4	2	2	3
US2856794	1955	2	4	2	2	3
US2914967	1956	2	4	2	2	3
US3287995A	1966	2	4	2	2	3
US3354752A	1967	2	4	2	2	3
US3541886A	1970	2	4	2	2	2
US4357840A	1982	3	6	0	2	3
US4884472A	1989	2	4	2	2	3
US8512198	2013	2	4	2	2	4

Since only one patented four-speed design exists with complexity 8, this data motivated the generation of novel four-speed transmission designs using two single planetary gears, two clutches and two brakes. The results of this generation process are presented in Section 6 and show that this methodology is able to generate novel designs with maximized capability.

The following Sections 4 and 5 will describe the methodology in detail.

4. MODEL-BASED TRANSMISSION DESIGN

The presented methodology requires the designer's input solely during the creation of the initial declarative design model to implicitly define the set of automatic transmissions. This model holds the information on the transmission components and the designers intent and expertise.

The transmission components are modeled as blocks with ports that define their connectivity to other components. A detailed representation of the modeling language is given by Nicolai et al. ⁽²²⁾. In this work only necessary explanations of parts of the language will be given.

Figures 2 and 4 show an example of the used declarative design model. The presented system can be used for the generation of two-stage automatic transmission architectures, where one motor is connected on a driven axle through a gearbox. The *gearbox* is composed of two *planetary gears* (blue), two *clutches* (green) and

two *brakes* (orange). The modeling language allows the system to be altered in order to cover a larger amount of designs. This however widens the design space significantly and does not suit the purpose of the presenting this research.

Fig. 2 Declarative design model for automatic transmissions.

Figure 3 shows that the transmission components were modeled with stick diagram representations in mind. Additionally it is important to mention that the *planetary gears* and *clutches* (as shown in Figure 2) have optional ports (ports with a dotted border). Optional ports allow for more variability in the topology generation as these do not have to be used in the generated solutions. In this example all ports are of the same type (red) and thus span a large amount of possible connections.

Fig. 3 Transmission components visualized by stick diagrams.

from left to right: *planetary gear*, *clutch*, *brake*

The developed graphical modeling language implicitly defines a set of rules for the generation of topologies. Any instance of a component is not allowed to connect to itself (unless specified otherwise by the designer), e.g. a *clutch* cannot connect to itself (creating a closed loop in this case), it can however be connected to other *clutches*.

The declarative design model used in this research has additional constraints that represent engineering knowledge and the designer's intent. Figure 4 shows the visual representation of these constraints. Three different types of constraints are used: Prescriptive connections (P1), limiting connections (L1) and restrictive connections (R1-R5). The concepts behind these constraint types is explained in detail by Nicolai et al. ⁽²²⁾ and is described here only briefly.

Constraint P1 represents the designer's choice to enforce system designs that connect the input shaft (i.e. the left port of the

gearbox component) directly to a *clutch*. It is noteworthy, that P1 avoids the generation of some of the patented transmission designs, where the input shaft is directly connected to a *planetary gear*.

Fig. 4 Internal Block Diagram of the Gearbox with Components and Additional Constraints P1, L1 and R1-R5.

Constraint L1 limits the number of connections between any two *planetary gears*. An upper limit of 2 (not shown in Figure 4) avoids two *planetary gears* to be connected directly by three or more shafts.

Constraint R1 forbids any *brake* to be connected to a port of the *gearbox* (i.e. the input or output shaft).

Constraint R2 forbids any two *brakes* to be connected to each other as they could not be connected to the system in that case. This constraint is not necessary per se as the generation process avoids unconnected solutions. It is used here, because it does accelerate the architecture creation process slightly, as a set of unconnected solutions is not generated and does not have to be filtered out.

Constraint R3 forbids any two *clutches* to be connected by their bottom ports. The intent behind this constraint is to avoid the creation of a simple shaft that has to be engaged by two shifting elements in order to be active. The same intent stands behind constraint R4, which forbids any *brake* to be connected the bottom port of any *clutch*.

Constraint R5 forbids the output shaft to be connected to the bottom port of any *clutch*, as this connection would allow a single *clutch* to completely disengage the driven axle from the transmission system.

The presented system components can be further defined with parameters, e.g. cost, weight, size or other user-defined parameters. In this case the *planetary gear* components are further defined by the planetary gear ratios as design variables. The planetary gear ratio defines the ratio between the number of teeth on the ring gear to the number of teeth on the sun gear $n = N_R/N_S$. Section 5.3. describes how these design parameters are used to optimize the transmission architectures.

The following section will explain how this declarative design model is used for the generation of topologies and the evaluation and optimization of the topologies for the creation of transmission architectures.

5. ARCHITECTURE GENERATION AND OPTIMIZATION

The architecture generation process is initiated by the definition of the declarative design model. The topology generation process (Section 5.1) defines a numerical constraint problem based on the central design model, where the solutions are numerical representations of the generated topologies. These topologies are evaluated based on transmission-specific features (Section 5.2) such that invalid and infeasible solutions are discarded. This is followed by the optimization of the design parameters (Section 5.3) since the quality of each design should be evaluated based on its optimal configuration.

5.1. Topology Generation

For the topology generation a constraint problem is set up, where the solutions are numerical representations of individual topologies. The solving of the constraint problem aims to find a set of connections between components, which results in a connected system. This process is constrained by various constraints inherent to the modeling language.

As the problem is solved, new constraints are added which avoid each topology to be generated again. Thus the problem is solved and further defined iteratively, until the process is stopped or the solution space is empty ⁽²²⁾.

As previously mentioned the solutions define which connections compose a connected system. Based on this solution each system (i.e. topology) is instantiated and can be evaluated for its viability as the architecture of an automatic automotive transmission.

5.2. Evaluation

The evaluation of the potential transmission architectures aims to determine the quality of each generated topology, such that the best candidates can be used for further development. The aim of the presented methodology is to determine the quality of each design based on automatically generated simulation results.

However simulations can be expensive compared to quick numerical evaluations. In order to reduce the cost of the evaluation by simulation, two transmission-specific analyses are executed first. These determine the manufacturability and feasibility for the use in automobiles of each generated transmission topology.

These analyses allow a set of topologies to be discarded before simulation, if those solutions are invalid or unfeasible.

The first analysis uses graph based planarity detection in order to determine whether a topology can be constructed in physical space. For this a transmission topology is translated into a graph

representation, where edges describe the physical connections between the planetary gears, clutches, brakes and the gearbox housing (i.e. input shaft, output shaft and the fixed casing). In this analysis a planar graph represents the manufacturability of the transmission design.

The second analysis employs the matrix method in order to determine the number of available transmission ratios. A matrix A is defined by the governing equations of the planetary gears, the connections between all components and the connections created by the engaged shifting elements $s_{engaged}$. The right hand side vector b is defined by the speed of the input shaft ω_{in} . Finally the system

$$A(s_{engaged})x = b(\omega_{in})$$

can be solved to compute the vector x which is defined by the rotational velocities ω of all components.

In order to determine the number of available transmission ratios, first all possible combinations of engaged shifting elements are enumerated. Then for each combination a matrix A and vector b are defined. For each solution x the rotational velocity of the output shaft ω_{out} is read and the transmission ratio

$$i = \omega_{in}/\omega_{out}$$

is computed.

As this is done for all possible combinations of engaged shifting elements, all available transmission ratios are computed. Thus the total number of forward and reverse speeds of each transmission architecture can be determined.

This can be used to discard topologies that do not support a user-defined number of necessary forward and reverse speeds.

After the execution of this analysis the methodology has produced a set of valid and feasible transmission architectures for an automotive application.

5.3. Optimization

In order to determine the best candidates from the generated transmission architectures, an optimization process is applied to each design. The optimization aims to minimize an objective function J by changing the design parameters. In this application the planetary gear ratios n_l are the only design parameters. A planetary gear ratio n_l is defined as floating point numbers and is not necessarily the ratio of two numbers of gear teeth, but can be approximated by them.

The optimization aims to find optimal values for the planetary gear ratios n , such that four-speed configurations are generated. For this it is assumed that the optimal gear step is $\varphi_{opt} = 1.4$, the optimal total range of transmission ratios is $r_{opt} = \varphi_{opt}^4 = 3.8416$, the optimal transmission ratio of the highest gear is

$i_{4,opt} = 1.0$ and the optimal transmission ratio of the lowest gears is $i_{1,opt} = r_{opt}/i_{4,opt} = 3.8416$.

Two objective functions J_φ and J_n are defined as follows:

$$J_\varphi = \max_k (1 - \varphi_k/\varphi_{opt})$$

and

$$J_n = \max_l S(n_l)$$

where $\varphi_k = i_{k+1}/i_k$ is the k -th gear step, i_k is the transmission ratio of the k -th gear, n_l is the planetary gear ratio of the l -th planetary gear and $S(n)$ is a function that evaluates a planetary gear ratio n based on the corresponding mean service life expectancy, according to the investigation of planetary gears with three planet gears by Savage et al. ⁽²³⁾. The function $S(n)$ returns a higher value for lower mean service life expectancy.

Further the optimization is restricted by the following inequality constraints:

$$1.5 \leq n \leq 10.0$$

$$N_{fwd} \geq 4$$

$$N_{rev} \geq 1$$

$$\varphi_k \geq 1.05$$

$$(1 - |i_{rev}/i_{1,opt}|) \leq 0.5$$

$$(1 - |i_1/i_{1,opt}|) \leq 0.5$$

$$(1 - |r/r_{opt}|) \leq 0.5$$

where N_{fwd} is the number of forward speeds, N_{rev} is the number of reverse speeds. The first constraint defines the boundaries of the optimization parameters. The second and third assert that the generated designs support the wanted amount of forward and reverse speeds. The fourth asserts that no transmission ratio is redundant. The last three constraints limit the variation of the reverse gear, lowest gear and total range from their optimal values to 50%.

In order to create a pareto front of optimal design parameters for each generated transmission design the following function $J(\alpha)$ is minimized for a range of values for $\alpha \in [0,1]$ such that all constraints are considered.

$$J(\alpha) = \alpha J_\varphi + (1 - \alpha) J_n$$

In summary the optimization aims to minimize the worst gear shift behavior (expressed through J_φ) and maximize the minimum mean service life of the planetary gears (expressed through J_n) while constraining the resulting transmission ratios to an acceptable range (expressed through constraints). This optimization has been executed for each of the generated four-speed designs and the results are presented in Section 6.

Current development however aims to optimize the design parameters based on the results of drivecycle simulations such that fuel consumption is minimized. This requires automatically

generated simulation models for all generated architectures and is further discussed in Section 7.

6. GENERATED TRANSMISSION ARCHITECTURES

As described in Section 1 the presented methodology was applied to the automated design of transmission architectures composed of two planetary gears, two clutches and two brakes. The generation of architectures was guided by the declarative design model described in Section 4.

Fig. 5 Schematic representations of three generated four-speed automatic transmission designs with a possible set of transmission ratios. These designs are composed of two planetary gears, two clutches (A, B) and two brakes (C, D) each.

In this application 3.750.000 (non-unique) transmission topologies were generated in 168 hours until the process was stopped by the user. The exact number of possible unique transmission designs is unknown for this application. For each transmission topology a manufacturability analysis was executed, followed by a kinematic analysis. The kinematic analysis determines how many forward and reverse gears each transmission topology supports and optimizes the transmission ratios (as described in Section 5.2 and 5.3).

For increased efficiency of the generation process all transmission designs with less than four forward speeds and one reverse speed were immediately discarded.

Figure 5 shows the three generated four-speed transmission designs. Of these three designs only one is presented in a patent to the knowledge of the authors: Design c) is presented in the US8512198 patent.

In an effort to further compare the generated designs to each other Figure 6 shows the pareto fronts that resulted from the optimization processes for the generated designs four-speed designs. Note that the x-axis is inverted and the maximum gear step sizing error is smaller to the right.

Design b) has a pareto front that consist only of a single point. This can be explained as this design is only able to conform to all constraints with a single set of planetary gear ratios.

Fig. 6 Pareto fronts of the generated transmission designs a) - c) showing minimum planetary gear mean service life in hours over the maximum gear step sizing error in percent. The optimal gear step is defined as $\varphi_{opt} = 1.4$. Square points are represent the data shown in Figure 5.

In Figure 6 it can be seen that the novel four-speed design a) has a longer mean service life of their planetary gears and gear steps that are closer to the optimal gear step size than the other designs. This proves that the presented methodology is able to generate designs that are superior to the patented transmission design c) in terms of achievable gear step quality and mean service life.

The exact values of these transmission designs may be improved in future development by a larger optimization loop that includes the evaluation of fuel economy. In order to evaluate the generated designs in terms of fuel economy a simulation is necessary. The following section will present the current efforts towards automatic fuel economy simulations.

7. TOWARDS AUTOMATICALLY GENERATED SIMULATION MODELS

Such that the generated transmission designs can be evaluated through simulation, each generated transmission topology and its optimized planetary gear ratios have to be represented by a simulation model.

In this application the transmission designs were simulated in LMS Imagine.Lab Amesim. For this a set of simulation components have to be selected from existing libraries. These components have to be connected such that a generated transmission topology is represented and the parameters have to be adjusted according to the optimized planetary gear ratios.

To support this process for each generated transmission architecture, the components of the initial declarative design model are augmented with simulation models as described in Section 7.1. This model augmentation is then used to generate and execute the simulation for each generated transmission architecture as described in Section 7.2.

7.1. Model Augmentation

Each component of the declarative design model is initially only defined by its ports and thus implicitly by the possible connections to other components. In order to support the simulation of the generated architectures the component are augmented with simulation models. The simulation model augmentation allows each component to represent a function or behavior.

By connecting the simulation models according to the connections of each generated architecture a fully connected simulation can be generated and executed. The simulation results can then be used to evaluate each generated architecture ⁽²⁴⁾.

The augmentation of the components with simulation models is part of the initial creation of the declarative design model. The ports of each component have to be mapped onto the ports of its simulation model. Further some components may represent entire subsystems of simulation models, which have to be created.

For the design of automatic transmissions the planetary gear and clutch components have been augmented with multiple simulation models. As these components have optional ports (i.e. ports that may not be connected in an architecture) they are augmented with multiple simulation models that represent the varying usages of the optional ports. It is important to note, that both simulation models employ the same governing equations during simulation.

Further the mapping of the system component models onto the simulation models includes parameters such as the planetary gear ratio of each planetary gear. For the generation and kinematic analysis of transmission architectures the planetary gear ratios is defined as a single value. In the simulation environment this is however represented by the number of teeth on the ring and the sun gear. Thus the value is converted by fixing the number of ring gear teeth and setting the number of sun gear teeth to a certain value accordingly, while minimizing the conversion error.

7.2. Generation and Execution of Simulations

At the current state of development the generation of simulations is not fully automated yet. The process involves the instantiation of the component augmentations in the simulation environment. This is followed by the instantiation of the connections between the simulation models according to each generated architecture.

Further the parameters of the simulation model are set according to the parameters determined during the generation of the transmission architectures. This data includes the shifting patterns for each gear of the transmission design, which is determined during the kinematic analysis of the transmission topology.

Because the generated simulation models only represent the engine, transmission and driven axle, a World-wide harmonized Light duty Test Cycle (WLTC) simulation cannot be executed yet. For this a controller system for control of acceleration, deceleration and optimal changing of gears (which is not considered during the generation of transmission architectures) is added to the simulation such that the WLTC is followed.

Finally the simulation results shall be analyzed for the fuel economy of the generated transmission designs, such that the best designs can be chosen for further product development.

8. CONCLUSION

This paper presented a computational design synthesis approach for the generation of novel automatic transmission designs for automobiles. An analysis of 1022 patented automatic transmission designs is presented in order to compare the generated transmission designs to existing designs.

The application shows the generation of three four-speed transmission designs, of which three are novel designs. These three designs are optimized and compared to each other in terms of minimum planetary gear mean service life and the quality of their achievable gear steps. Thus it is shown that the presented methodology allows the designer to generate novel transmission designs and evaluate the designs automatically. As the generated transmission are only evaluated based on their planetary gear ratios and transmission ratios, an approach towards automatic simulation is introduced.

Future development aims to not only evaluate the designs based on fuel economy simulations but also optimize to optimize the planetary gear ratios based on the simulation results. Furthermore the models of the transmission design components could be extended with gear contact models that in addition allow an automatic evaluation of dynamic effects and durability.

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