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Comparative Study of Dynamic Characteristics of Automobiles Using Mechanical Transmission and Hydraulic Torque Converter

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ABSTRACT; This paper presents a comparative study of the dynamic characteristics between the two main automotive driveline systems: mechanical transmission (MT) and torque converter automatic transmission (AT). Through a combination of theoretical calculations and experimental simulation, with characteristic indicators such as tractive force, maximum speed, gradeability, dynamic factor, and maximum acceleration, the paper highlights the distinct advantages of each transmission system under real-world operating conditions. The research results emphasize the role of the hydraulic torque converter in improving starting performance, acceleration, heavy-load operation, and grade climbing, while also clarifying the advantages of transmission efficiency and direct control of mechanical transmissions. These analyses contribute scientific evidence for selecting and optimizing appropriate driveline configurations for each vehicle type and exploitation objective in modern automotive engineering.

Keywords: Automotive dynamics, mechanical transmission, torque converter automatic transmission, hydraulic torque converter, tractive force, gradeability, powertrain efficiency, comparative analysis, vehicle performance, torque multiplication, dynamic characteristics, driveline configuration.

I. INTRODUCTION

The driveline is one of the key components determining operational efficiency, flexibility, and user experience in modern automobiles. Mechanical transmission (MT) is notable for its high transmission efficiency, simple structure, low maintenance cost, and immediate driver feedback. In contrast, the torque converter automatic transmission (AT) excels due to its ability to automatically amplify torque, provide smooth operation, and optimize performance during starting, acceleration, as well as heavy-load and grade driving. The choice between these two solutions directly affects fuel consumption, acceleration, gradeability, and suitability for specific operating conditions [1][2] .

II. DYNAMIC CHARACTERISTICS OF AUTOMOBILES USING MECHANICAL TRANSMISSION AND TORQUE CONVERTER

2.1. Overview of International Research on Mechanical Transmission and Torque Converter

- Hydraulic torque converter driveline systems not only improve the operation of conventional vehicles but also play an important role in hybrid drives and electric vehicles [1]
- The trend towards permanent magnet synchronous motors (PMSM) or synchronous reluctance motors (SynRM) allows automatic driveline systems (with torque converter or electronic clutch) to maximize efficiency, reduce energy consumption, and increase system longevity [2]
- Modern works employ digital simulation, machine learning, and intelligent control to optimize driveline management, thereby improving traction, reducing fuel consumption, and emissions [3]

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2.2. Analysis and Comparison of Dynamic Characteristics

2.2.1. Engine External Characteristics

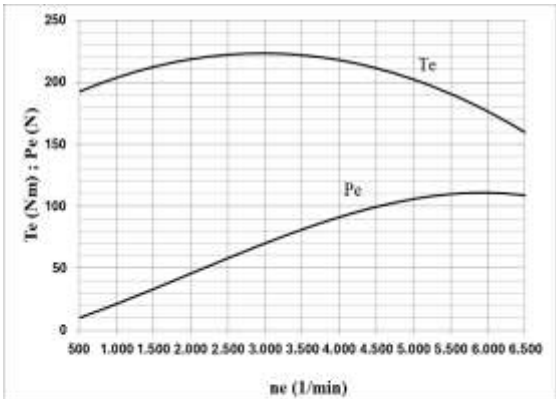

Fig.1. Engine torque and power curve versus engine speed

- The torque curve (Te) reaches its peak at ne = 3000 rpm (223.47 Nm), then gradually decreases as engine speed increases.
- The power curve (P_e) peaks at n_e = 5930 rpm (111 kW) before declining due to the drop in torque.
- Remark: The engine has a wide effective operating range, favorable for both low-speed (starting, climbing) and high-speed operation [4].

2.2.2. Maximum Tractive Force Curve

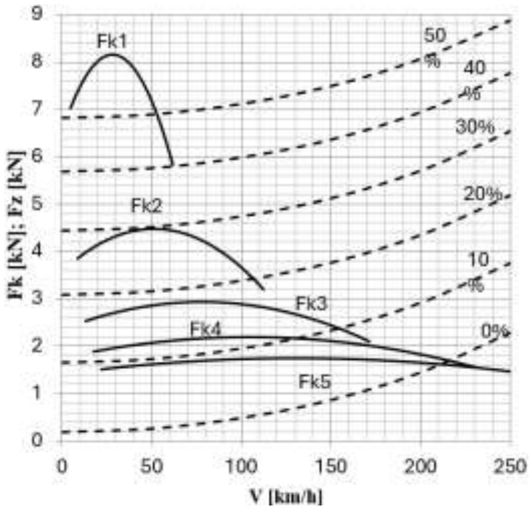

Fig. 2. Comparison of maximum tractive force among gear ratios

Acceleration and gradeability at different gear ratios must be examined. In the tractive force diagram (Fig. 3.2), the actual tractive force at each gear and the required tractive force at different grades are presented as functions of vehicle speed. The actual tractive force is reduced by the transmission efficiency ηt, due to losses in auxiliary devices such as the pump.

From the straight-line tractive characteristic, we can determine:

- The maximum tractive force of the vehicle in each gear (first gear: 8158 N).

III. BASIC PARAMETERS OF THE TORQUE CONVERTER:

3.1. Torque Converter Ratio:

$$K_{bm} = \frac{M_T}{M_B} \quad (1)$$

Where: M_T: Torque on turbine shaft; M_B: Torque on pump impeller shaft .

The torque converter ratio depends on vehicle operating conditions. When resistance increases, vehicle speed decreases, causing the turbine shaft speed to drop, which increases MT and thus K_{bm}. Kbm reaches its maximum when the turbine is completely stalled (nT = 0). Conversely, when resistance decreases and vehicle speed increases, the torque converter ratio reduces correspondingly. The automatic torque multiplication characteristic of the hydraulic torque converter is due to the changing fluid flow impact on the turbine blades as rotational speed varies [5].

3.2. Torque Converter Speed Ratio

The speed ratio (ibm) is the ratio of turbine shaft speed nT to pump impeller speed nB:

$$ibm = \frac{nT}{nB} \quad (2)$$

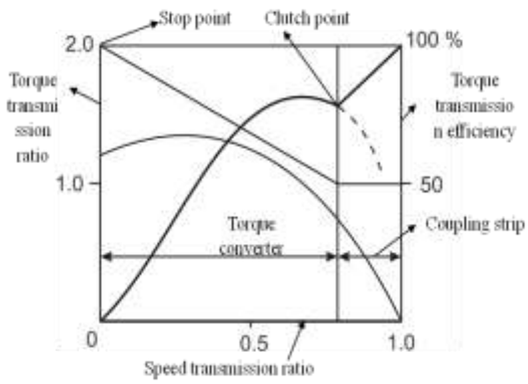

Fig. 3. Stall and Clutch Points

Torque converter operation is divided into two ranges:

- Torque multiplication range, where torque amplification occurs.
- Coupling range, where torque is transmitted without amplification. The clutch point separates these two regimes.

3.3. Torque Converter Efficiency

The efficiency of the torque converter indicates how effectively the energy transferred to the pump impeller is delivered to the turbine. As torque is transmitted at a nearly 1:1 ratio in the hydraulic coupling range, efficiency is maximized.

3.3.1. Calculation and Selection of Hydraulic Torque Converter:

The outer diameter of the torque converter working chamber is determined by the formula:

$$D_a = \sqrt[5]{\frac{T_1}{\lambda \cdot \gamma \cdot n_P^2}} \quad (3)$$

Where:

T_1: Input torque to the torque converter, here $T_1 = T_{emax}$ = 223.47 Nm = 22.347 kgm, the engine's maximum output torque.

λ : Torque coefficient of the converter;

γ : Specific weight of the torque converter fluid, here ATF DEXRONII,

γ = 853 kg/m^3 = 8530 N/m^3.

Remark: Based on the vehicle type, engine power and torque, and by referencing similar vehicles, a sensitive-type converter with a dimensionless characteristic curve as above is selected.

3.3.2. Selected Parameters of the Hydraulic Torque Converter:

- Sensitivity:

$$\pi = \frac{\lambda_1}{\lambda_2} = \frac{3.2}{0.8} = 4 \quad (4)$$

- Outer diameter: Da = 247 mm
- Torque transmission power limit:

N = 20–60 Hp

- Maximum efficiency: $\eta_{max} = 0.9$
- Fluid specific weight: γ = 853 kg/m^3 = 8530 N/m^3
- Maximum torque multiplication factor: $K_{max} = 2.8$

3.2.3. Construction of the Torque Converter Input Characteristic Curve:

The input characteristic curve reflects the relationship between the torque on the pump impeller shaft (T_1) and its rotational speed (n_1), with torque coefficients varying by speed ratio i.

The relationship can be expressed as:

$$T_1 = \lambda_1 \cdot \gamma \cdot n^2 \cdot D^5 \quad (5)$$

For sensitive mixed-type converters, as the torque coefficient changes, T_1 is determined by the respective λ1 values. Using the dimensionless characteristic curve of the converter, for each speed ratio ibm, λ1 is identified and corresponding torque values at different input speeds ($n_1 = n_e$) are calculated[5].

By substituting the values into formula (5), we have:

At i=0, $\lambda1\_0 = 3.22\times10^{-6} \rightarrow T1\text{-}0 = \lambda1\_0 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.1, $\lambda1\_0.1 = 2.93\times10^{-6} \rightarrow T1\text{-}0.1 = \lambda1\_0.1 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.2, $\lambda1\_0.2 = 2.67\times10^{-6} \rightarrow T1\text{-}0.2 = \lambda1\_0.2 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.3, $\lambda1\_0.3 = 2.4\times10^{-6} \rightarrow T1\text{-}0.3 = \lambda1\_0.3 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.4, $\lambda1\_0.4 = 2.13\times10^{-6} \rightarrow T1\text{-}0.4 = \lambda1\_0.4 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.5, $\lambda1\_0.5 = 1.87\times10^{-6} \rightarrow T1\text{-}0.5 = \lambda1\_0.5 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.6, $\lambda1\_0.6 = 1.6\times10^{-6} \rightarrow T1\text{-}0.6 = \lambda1\_0.6 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.7, $\lambda1\_0.7 = 1.33\times10^{-6} \rightarrow T1\text{-}0.7 = \lambda1\_0.7 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.8, $\lambda1\_0.8 = 1.07\times10^{-6} \rightarrow T1\text{-}0.8 = \lambda1\_0.8 \cdot \gamma \cdot n^2 \cdot D^5$

At i=0.9, $\lambda1\_0.9 = 0.8\times10^{-6} \rightarrow T1\text{-}0.9 = \lambda1\_0.9 \cdot \gamma \cdot n^2 \cdot D^5$

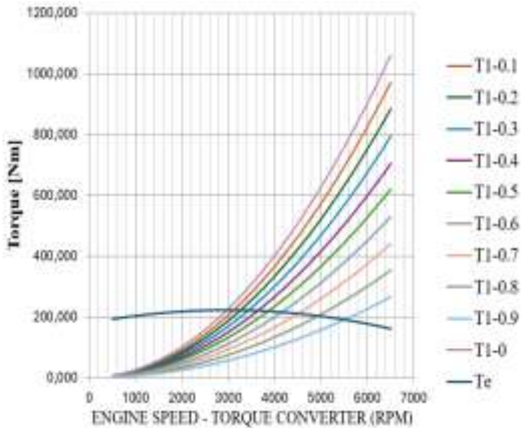

Fig. 4. Input characteristic curve of the torque converter

This curve shows the relationship between input torque (T1) and pump impeller speed (n1) for varying torque coefficients by speed ratio i. As i decreases (lower turbine speed relative to the pump), the torque coefficient increases, resulting in higher T1, illustrating the torque multiplication advantage during starting or heavy load. Conversely, as i approaches 1 (turbine speed approaches pump speed), the torque coefficient drops and T1 decreases, indicating a transition to

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coupling mode with no further torque multiplication.

3.2.4. Construction of the Torque Converter Output Characteristic Curve:

The output characteristic, representing the combined effect of engine and converter, is used for tractive calculations.

From intersections at Ai, values of T1, n1 corresponding to selected speed ratios i are determined. According to the dimensionless characteristic curve, for each i, values of T1, n1, and η are found, from which the following are calculated:

$$n_2 = i \cdot n_1^* \quad (6)$$

$$T_2 = T_1^* \cdot K_{bm} \quad (7)$$

$$P_1 = T_1^* \cdot n_1^* \quad (8)$$

$$P_2 = P_1 \cdot \eta \quad (9)$$

Using these values, a table and the output characteristic curve are established

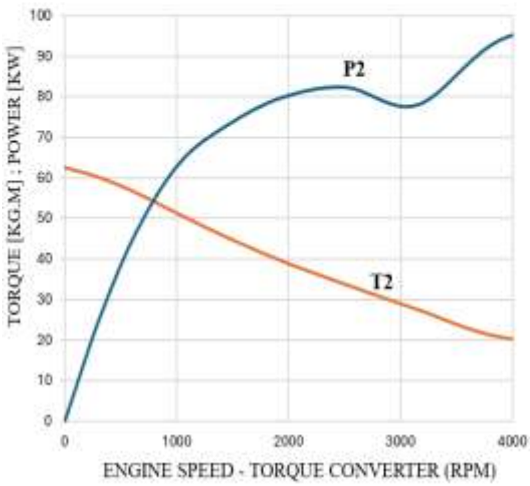

Fig. 5. Characteristics graph on the output shaft of the torque converter

For the torque converter output characteristic curve, changes in turbine shaft torque and power (T2, P2) are observed as speed ratio i varies. At the initial stage (low i), output torque is high, matching tractive requirements during starting or hill climbing. As i increases, output torque decreases and power peaks in the midrange before slightly dropping as i approaches 1, reflecting the converter’s automatic adjustment for optimal tractive force at each operating stage.

IV. CONCLUSION:

From the analysis of the input and output characteristic curves, it is clear that the hydraulic torque converter is an effective intermediary for managing and amplifying torque between the engine and the drivetrain. The converter enables smooth starting, strong acceleration, and easy grade climbing by automatically varying the speed ratio and torque multiplication as needed. At high speeds, the converter transitions to coupling mode, reducing losses and improving efficiency.

Compared to mechanical transmission, a system with a hydraulic torque converter offers smoother operation, reduces driver workload, and optimizes transmission performance across a wide range of speeds and loads. However, at light loads, the converter’s overall efficiency may be lower due to inherent energy losses. Thus, driveline configuration should be selected based on specific usage requirements.

The results clarify the important role of the hydraulic torque converter in modern automatic drivetrains, especially in automatic vehicles, heavy trucks, and specialized vehicles where flexible and reliable operation is essential.

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