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HYBRID PROPULSION SYSTEM WITH A GYRO COMPONENT FOR ECONOMIC AND DYNAMIC OPERATION

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16. Abstract
The design of a hybrid drive with gyro components is described and its drive components for a medium class private car are discussed. The arrangement of the drive elements is based on the stringent FCCI operational requirements. By reason of its high specific output the gyro component affects the short-period output of the drive by accelerating and slowing down and -- because of the mechanical transfer of kinetic energy between the gyro and the vehicle -- it affects also the energy balance in the case of intermittent operation. Energy can be taken in as desired either in the form of fuel or as fuel and current. A high energy recovery efficiency as well as the favorable operating range of the I.C. engine makes it possible to reduce the fuel consumption per unit distance travelled to almost half that for a private car with a traditional IC. engine.

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Introduction /5

In congested areas vehicles are increasingly being used in a markedly stop-and-go manner. In the future their maximum speed will be increasingly limited -- even on highways and freeways. The motive energy -- in particular, fossil fuel -- is becoming more expensive.

Given these changing limit conditions the following properties of vehicle drive mechanisms are gaining in importance:

1. high acceleration with reduced cruising power;
2. high propulsive efficiency, in particular in the partial throttle region;
3. regenerative braking with high recovery efficiency;
4. low pollutant emission and
5. low noise production.

These requirements are not met -- or only unsatisfactorily so -- by traditional internal combustion engines.

Limits to electric power units -- as a potential alternative to the internal combustion engine -- are set by the small energy density and power density of electric storage batteries, by a small recovery efficiency and by presently high production costs [1]. Hybrid propulsions systems with an electric motor and internal combustion engine as well as an electric power addition [2] proved to be too heavy and too expensive, at least for passenger vehicles. With respect to the above requirements a hybrid propulsion system was designed with an additional gyro

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* Numbers in the margin indicate pagination in the foreign offprint text.

component (flywheel) and with a mechanical power addition [3]. This is described below and discussed with respect to its properties for use as a propulsion system for passenger vehicles.

2. Hybrid Propulsion System with a Gyro Component

2.1. Structure of the Propulsion System

In contrast to suggestions which have been made [4,5], the propulsion system consists of each of the following: a gyro component (1) an electric component (2) and an internal combustion engine component (3).[1] The outputs of these components are mechanically overlapped in a structure as shown in Fig. 1.

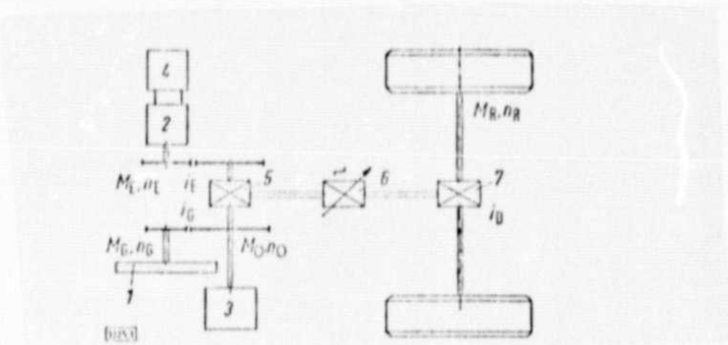

Fig. 1. Structure of a hybrid propulsion system with a gyro component.

1 Gyro component
2 Electric propulsion system
3 Internal combustion engine
4 Battery
5 Differential gearing
6 Shift gears
7 Differential gearing for the drive shafts

The electric motor (2) is connected via a transmission i_e with an input of the differential gear 5. The internal combustion engine 3 is couppled with a second input and the gyro component 1

1. In what follows, in so far as the distinction is necessary, this hybrid propulsion system will be referred to in abbreviated form as the G.E.I propulsion system.

by means of transmission i_G. The output of this differential gear 5 is connected by means of a shift gear 6 with the usual differential gear 7 of the drive shaft.

2.2 Function of the Propulsion System

In the case of stationary operation the positive output of the i.c. engine 3 with the positive or negative output of the electro machine 2 (motor or generator operation) overlap in differential gear 5. The resulting sum or difference output is transmitted via shift gear 6 and drive shaft differential 7 to the drive gears. The gyro component 1 runs with constant rpm, i.e. without any reduction in power. In the case of stationary braking the flow of power is reversed.

With a standing vehicle the electric motor and i.c. engine stand still or rotate in opposition (battery charging). /6

In order to accelerate the moment M_E of the electric motor is increased by activating the foot pedal (gas pedal). Because of the balance of moments on differential gear 5 this produces the moment $i_E M_E$ on its inputs and outputs. On the one hand, this operates as a driving moment on shift gear 6 and also on the drive gears and on the other hand it operates on the parallel connection of the i.c. engine and gyro component. At full acceleration the moment $i_E M_E$ is one multiple greater than the moment M_0 of the i.c. engine. The large difference in moments $i_E M_E - M_0$ is thus supported by the gyro component and slows down the latter according to the following equation:

$$\Theta_G \dot{\omega}_G = (i_E M_E - M_0)/i_G \cdot \qquad (1)$$

In this connection the gyro component produces the following propulsive output:

$$P_G = \omega_G M_G = \omega_G (i_E M_E - M_0)/i_G \qquad (2)$$

Thus, with good efficiency, during acceleration kinetic energy from the gyro component is transmitted directly to the vehicle.

When decelerating this process takes place in the opposite direction. The gyro component is "recharged." A quantitative discussion of these relationships follows after the description and determination of the individual components.

If necessary, the drive can also be operated just electrically. In this case the required supporting moment of the differential gear 5 is applied on the i.c. engine side by means of a non-reversing lock.

3. Designing a Propulsion System for Passenger Vehicles

The components of the propulsion system are described and determined below in such a way so that a thus equipped passenger vehicle meets, as far as possible, the very demanding FCCI (U.S. Federal Clean Car Inititive) automotive performance figures for vehicles with low-polluting unconventional propulsion systems:

1. acceleration from zero to 100 km per hour in 16 seconds;
2. occasional maximum velocity of v = 136 km per hour;
3. travelling up an 8 km long 8% grade at a velocity of v = 64 km per hour and
4. a range of 320 km at a velocity of 112 km per hour.

3.1 Basic Vehicle

As a sample vehicle a 5-passenger car of a lower medium class European model with the following specifications is used as a basis:

1. weight when empty with G.E.I. hybrid propulsion system (this assumption is checked later on) m_{FO} = 1200 kg,
2. operating weight (including driver) for all the following tests m_{FB} = 1300 kg,

3. wheel resistance coefficient $f_R = 0.015$,
4. air resistance coefficient $c_W = 0.36$,
5. effective cross-sectional area $A = 1.8\ m^2$,
6. inertia moment of all four wheels $\Theta_R = 4.0\ kg\ m^2$.

The road resistances for this vehicle are shown in Fig. 2. Given the vehicle specifications the propulsion system components are now determined.

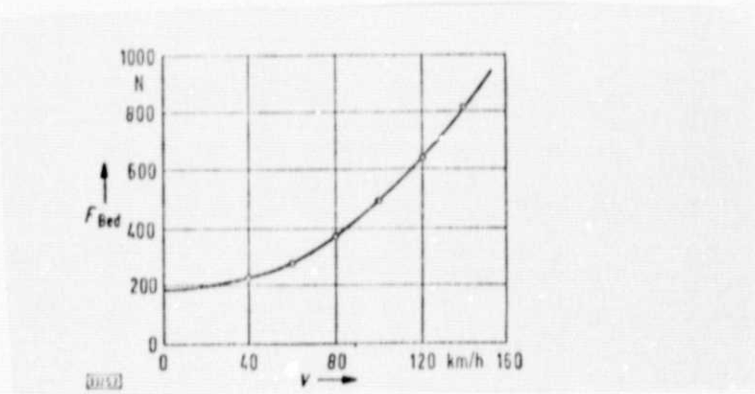

Fig. 2. Road resistance $F_{Bed} = F_R + F_L$ of the sample vehicle (passenger car) as a function of the velocity v of the car on a level plane.

3.2 Design of the Gyro Component

Basically the gyro drive is supposed to supply or store the energy for an acceleration or deceleration process. This does away with the problems which have to be overcome when designing the gyro drive as an actual automotive propulsion system [6-9]. Disregarding air and wheel resistance we thus get the following equation which relates the energy involved:

$$\Theta_G (\omega^2_{G\,max} - \omega^2_{G\,min})/2 = v'^2_{max}\, m_F/2 \qquad (3)$$

with Θ_G the inertia moment of the gyro component, Θ_{Gmax} the maximum angular velocity and ω_{Gmin} the minimum angular velocity of the gyro drive and v'_{max} the maximum velocity which should be reached in one go from a standing start. This maximum velocity is somewhat smaller than the steady maximum velocity v_{max}.

The inertia moment of the gyro drive and thus also its weight therefore decisively depends on the allowed maximum angular velocity ω_{Gmax}. Given the above mentioned form of the gyro drive, this is essentially limited by the permissible strain of the gyro material and by the fan losses. A suitable optimization by means of a computer program produced an easy-to-manufacture gyro drive made of 42 CrMo 4 with the following specifications: ω_{Gmax} = 1833 per second corresponding to n_{Gmax} = 17,500 U/min; Θ_G = 0.612 kg m^2 and thus with a diameter of d_G - 315 mm a mass of the gyro drive of m_G = 50 kg. /7

3.3 Design of the Internal Combustion Engine

The internal combustion engine is connected in parallel to the gyro drive by means of the fixed transmission i_G. Its output is basically determined by the requirement for a range of 320 km at a velocity of v = 112 km per hour. Because of the fairly small energy content of the electric battery (lead battery) the motive power of P_{F112} = 18 kW corresponding to 112 km per hour must nearly all be supplied by the i.c. engine.

Taking into consideration the transmission efficiency, we get the following:

$$P_O = 20 \text{ kW}.$$

In light of noise and vibration characteristics, weight, volume and costs, a single-disc Wankel engine was selected for the internal combustion engine. The characteristic curves for its maximum power moment and torque are shown in Fig. 3. The parallel hookup to the gyro drive makes possible a "stabilized," i.e. low emissions, control of the engine during load changes [10]. The transmission between the gyro drive and the Wankel engine turns out to be the following:

$$i_G = n_{max\,G}/n_{max\,O} = 3{,}47.$$

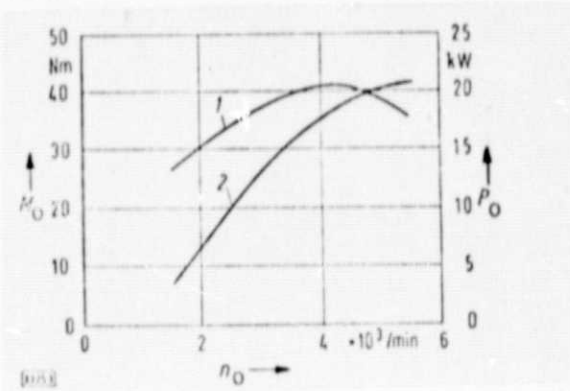

Fig. 3. Maximum torque and maximum power of the internal combustion engine as a function of rpm.
1 $M_0(n_0)$ 2 $P_0(n_0)$

3.4 Electric Motor and Battery Design

The electric components of the hybrid propulsion system (electric motor with power control and battery) are the largest in terms of weight and costs [11]. The minimum power to be supplied by the electric motor is given by the requirement for an occasional maximum velocity of 136 km per hour. The corresponding motive power (P_{F136} = 30 kW) is to be jointly supplied by the Wankel engine and the electric motor.

Allowing for the transmission efficiency, we get a cruising power of

$$P_E = 11 \text{ kW}.$$

A direct current shunt motor is selected for an operational voltage of 144 V. In conjunction with a suitable direct current regulator we get the measured performance graph shown in Fig. 4.

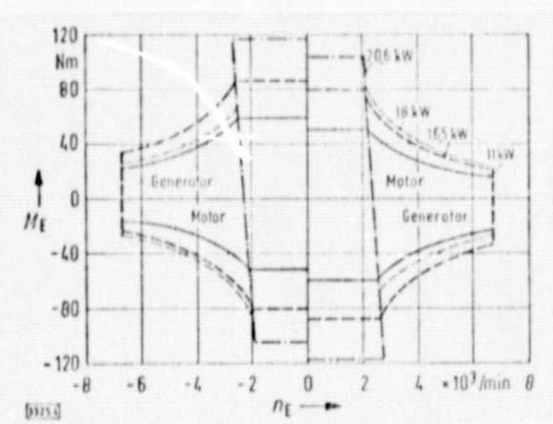

Fig. 4. Torque M_E (n_E) of the electric motor as a function of rpm for continuous and brief operation of 11 kW and 20.6 kW respectively.

The required battery size is determined primarily by the maximum power consumption of the electric motor which for brief periods (during acceleration processes) takes P_{Bmax} = 25 kW from the batter. For 12 starter batteries connected in series a maximum specific short-time output of 0.17 kW/kg is assumed [12]. This results in a battery mass of: m_B = 150 kg and, in the case of 1-hour discharging, an energy content of

$$E_{B\,1\,h} = 4{,}1\ \mathrm{kWh}.$$

For faster discharge rates, i.e. greater load, the available energy content decreases [13].

This relationship was measured for each of twelve 12-volt starter batteries of different manufacturers connected in series. The result is plotted in Fig. 5 for a battery weighing 150 kg. It is used as a basis below for steady energy and range considerations.

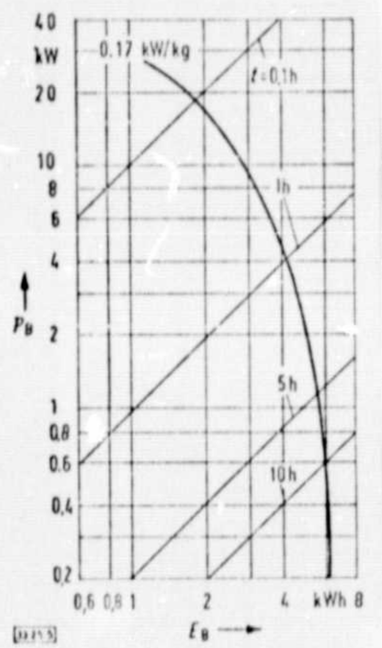

Fig. 5. Battery output as a function of energy removal with the discharge time t as the parameter for a specific output of 0.17 kW/kg.

In all steady operational conditions below v = 112 km per hour and for intermittent operation the battery can be selectively charged or discharged by suitably regulating the rpm of the Wankel engine, thus has no effect on the range.

3.5 Gear Design and Transmissions

The differential gearing for overlapping the outputs is constructed in the form of spur gear - planetary gearing because in this way the transmission i_E can be integrated into the differential and -- for adjustment purposes-- can be changed in a simple manner. For the differential gearing the following relationships apply in accordance with Fig. 1:

Balance of Moments

$$M_A = M_E\, i_E = M_0 + M_G/i_G. \qquad (4)$$

rpm addition

$$\omega_A = \omega_E/i_E + \omega_0. \tag{5}$$

Multiplying equations (4) and (5) results in the following output power at the differential:

$$P_A = \omega_A M_A = M_E(\omega_E + i_E \omega_0). \tag{6}$$

To achieve a high output power for a given power $P_E = M_E \omega_E$ of the electric motor a large i_E should be selected. A second criterion in the selection of i_E is the desire, for steady /8
operation conditions, to achieve a balance of moments of the electric motor and i.c. engine

$$i_E M_E = M_0$$

which is close to the rated speeds of these motors. Given the characteristic curves of voth motors from Figs. 3 and 4 i_E is chosen equal to 2. The total transmission in the lowest gear of the shift gear determines the maximum tractive force F_{z1} on the drive gears, hence the climb and acceleration capacity in first gear:

$$F_{Z1} = M_E i_D i_{S1} i_E / R_{dyn} \tag{7}$$

with i_D the transmission of the differential gear for the drive shaft, i_{S1} the transmission of the shift gear in first gear and R_{dyn} the dynamic wheel radius -- here 300 mm.

The total transmission in the highest gear results in an analogous manner from the road resistances at maximum velocity.

The maximum progression $i_{S,n}/i_{S,n+1}$ between two adjacent gears is given by the characteristic curve m_E (ω_E) of the electric motor if it is assumed that the take-off moment immediately before and after a shift should be equal in size (uniform acceleration).

Under these conditions two gears in the first instance prove to be sufficient with the following transmission ratios:

$$i_{S1} = 2{,}21, \quad i_{S2} = 1{,}0, \quad i_{SD} = 6{,}15.$$

Thus all the components and design specifications of the propulsion system are determined.

4. Propulsion System and Motive Outputs

The outputs of this hybrid propulsion system and the resultant motive output of the selected passenger car are considered jointly below.

4.1 Steady Driving Conditions

Steady driving conditions which are characterized by an inactive gyro component can -- for the time being ignoring the transmission losses -- be described by means of a nomogram which is based on the already known performance graphs of the components and of the vehicle (Figs. 2-4). This nomogram is shown in Fig. 6.

The performance graph of the electric motor is plotted in section (a) and that of the i.c. engine in section (c). In the middle section (b) is plotted the resistance graph of a vehicle with two gears. The distances on the two speed axes for the electric motor and the i.c. engine are divided in a ratio of i_E to the velocity axis. If one connects any two motor speeds n_E and n_O by a straight line, the intersection of this line with the velocity axis gives the accompanying velocity v of the

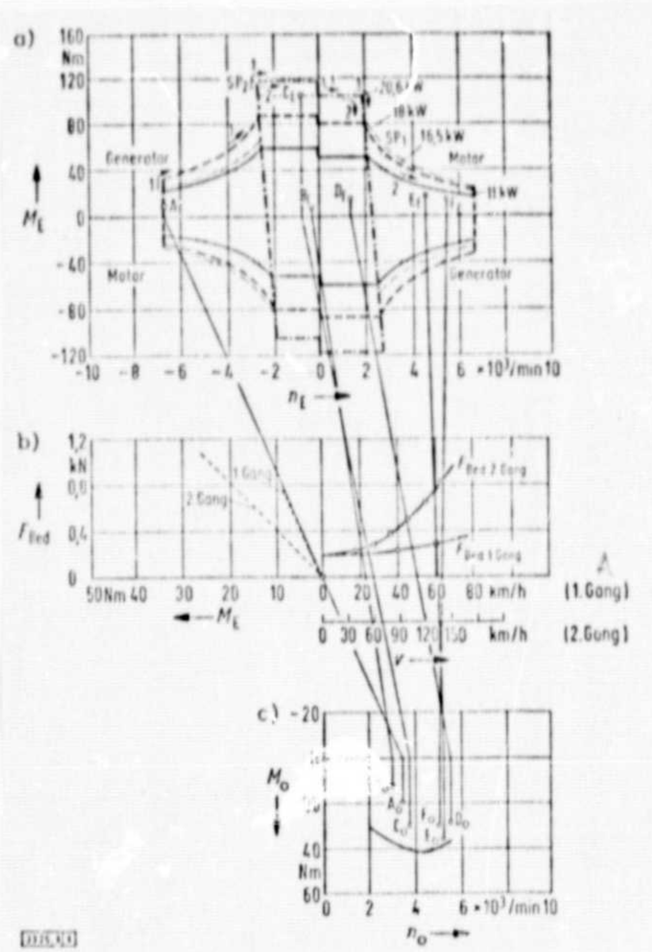

Fig. 6. Nomogram of the hybrid propulsion system for determining motove output under steady operating conditions. The graph plots the road resistances F_{Bed} over the velocity v of the passenger car as well as the torque M over the speed n of the electric motor and the i.c. engine
a) For the electric motor, 11kW under steady operating conditions and 2.6 kW for short period operation,
b) road resistances for a vehicle weighing 1300 kg,
c) for a 20 kW i.c. engine

Key: A) gear

vehicle in first or second gear (Fig. 6 b). The torques m_E and m_O corresponding to a given steady driving state are given by the corresponding road resistance F_{Bed} and gear (nomogram section b). For a few driving conditions the accompanying operating levels A-F are plotted in the two performance graphs. The points along line A characterize the charge of the battery in the case of a standing vehicle. The moment $M_E \triangleq A_E$ corresponds to an output of 7 kW. The points along line B correspond to a speed of v = 50 km per hour in second gear with the i.c. engine turning at n_O = 3000 rpm. Under these driving conditions the electric motor is still being operated in the generator range, i.e. the battery is charged with a small output.[1]

Also plotted in the nomogram are the FCCI driving output

1. The ideal speed of the Wankel engine n_{Oideal} is set by a regulator as a function of the charging state of the battery and external operating conditions. A description of the regulator is not given here.

requirements 2-4 for steady driving conditions (lines D, E and F):

FCCI requirement 2: occasional maximum velocity of v = 136 km per hour. This requirement is fulfilled in the nomogram by the operation characterized by line E. With respect to time it is not limited by the electric motor (long term output, 11 kW), but by the battery capacity. On the basis of the graph in Fig. 5 we get for v_{max} = 136 km per hour an operation time t_2 = 0.38 hours, i.e. a distance of s_2 = 51 km. /9

FCCI requirement 3: 8 km long 8% climb at a speed of v = 64 km per hour. To the air and wheel resistance at a speed of 64 km per hour is added in this case a climb resistance of 1020 N. The total resistance of 1320 N (beyond the range of the nomogram) is shown by the performance graph line F in first gear. The possible operating times or distances are again determined on the basis of Fig. 5 as t_3 = 0.41 hours and s_3 = 26 km respectively. Thus this requirement, i.e. 8 km, is easily fulfilled.

FCCI requirement 4: range of 320 km at a speed of v = 120 km per hour. This requirement is fulfilled by the operating conditions characterized in the nomogram by line D. After this distance either the tank must be filled or the battery charged or -- if the car is only refueled -- the motive output under steady operating conditions must be slightly reduced so that the battery can be recharged by the Wankel engine when the car is underway again.

Thus the FCCI motive output requirements under steady operating conditions are fulfilled.

4.2 Intermittent Driving Conditions

Intermittent driving conditions which are critically affected by the gyro drive can be illustrated only qualitatively in the nomogram.

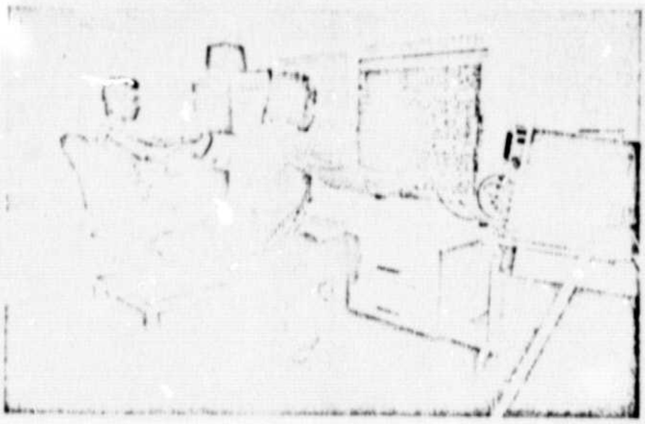

Fig. 7. View of the analog computer with computer console for propulsion system studies.

Thus, for example, when starting up with maximum acceleration -- beginning from the initial situation A -- the maximum torque curve for the electric motor will follow the line denoted by arrows and the number 1 in first gear. At n_E = 2800 rpm the car is shifted into second gear whereby the speed of the electric motor decreases to n_E = -2600 rpm (transition from shift point SP_1 to SP_2, and for the rest of the acceleration the maximum moment of the electric motor is available. During this process the "deficient" difference of moments $i_E M_E - M_O$ on the Wankel side of the differential is bolstered by the gyro drive and decreases its speed while it supplies a high gyro output.

In order to give a proper quantitative description of the various intermittent driving conditions and be able to simulated them in real time a model of the propulsion system and vehicle was formulated and reproduced on an analog computer. This model takes into consideration the efficiency or power losses of all the components in good approximation to the actual situation. Fig. 7 shows the computer with computer console.

To determine fuel consumption and pollutant emission of the Wankel engine the computer can also be operated on-line as a real component.

As examples of intermittent driving conditions acceleration events are first considered. For the case in which the speed of the Wankel engine at the beginning of the acceleration process is set fairly low at $n_{Oideals}$ = 2500 rpm (e.g. in city traffic) we get the velocity-time curve v(t) no. 6 as shown in Fig. 8 a. The increase in velocity, i.e. the acceleration, is at first not very high up to 10 km per hour because the electric motor has not yet reached the rpm range of its maximum moment $M_E(t)$ (curve 2). Above 10 km per hour the acceleration its maximum of 4 m/s^2. At v = 25 km per hour the shift is made into second gear.

At v = 52 km per hour we leave the speed range of the maximum moment $M_E(t)$ and the rpm curve $n_E(t)$ thus becomes flatter. At v = 86 km per hour, corresponding to a motor speed of n_E = 6700 rpm (curve 4), the acceleration has stopped. The speed of the i.c. engine has dropped from n_O = 2500 rpm to 1300 rpm (curve 5). The battery output $P_B(t)$ (curve 3) changes within the limits P_B = -22 kW

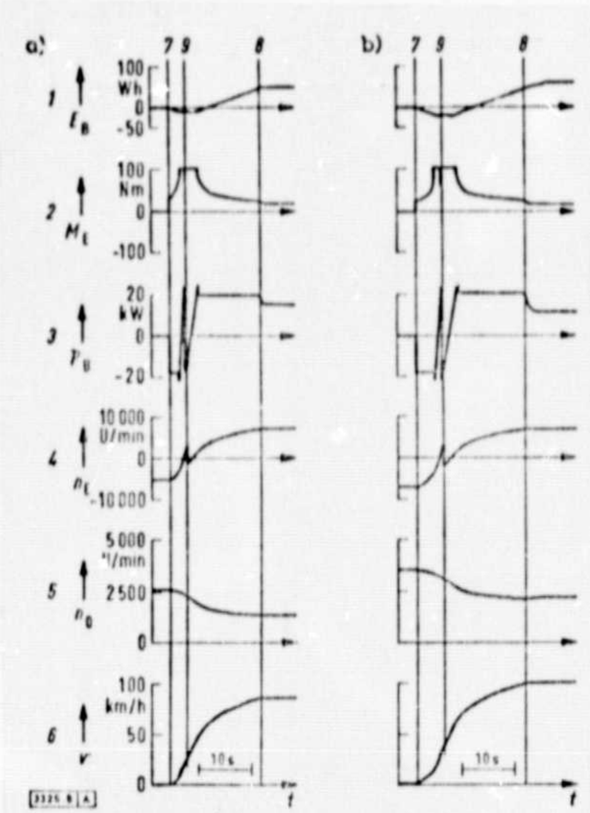

Fig. 8. Different events represented by an analog computer.
a) from 0 to 80 km per hour with an initial speed of the i.c. engine n_O = 2500 rpm,
b) from 0 to 100 km per hour with an initial speed of n_0 = 3350 rpm.
1. energy consumption (battery) $E_B(t)$
2. Torque (electric motor) $M_E(t)$
3. Output (battery) $P_B(t)$
4. rpm (electric motor) $n_E(t)$
5. rpm (i.c. engine) $n_O(t)$
6. Velocity v(t)
7. Start of acceleration
8. End of evaluation
9. Shift point

(generator operation) and P_B = +25 kW (motor operation). An amount of energy equal to E_B = 51 Wh is removed from the battery for the entire acceleration process.

Fig. 8b shows an acceleration process with an initial speed of the i.c. engine of n_{0ideal} = 3350 rpm. After 20 seconds a velocity of approximately 100 km per hour is reached. It is limited by the maximum speed of the electric motor. A further velocity increase -- maximum 136 km per hour -- can only be achieved by increasing the speed n_0 of the i.c. engine.

The FCCI requirement 1 of reaching a velocity of 100 km per hour in 16 seconds from a standing start is thus not fulfilled. This would require, for example, a third gear.

Fig. 9 shows curves in which all the events start with a steady velocity of v = 20 km per hour and a Wankel engine speed of n_{0ideal} = 5000 rpm. At v = 35 km the shift is made into second gear (curve 4). After 9.5 seconds a speed of v = 100 km per hour is reached, and only above v = 105 km per hour does /10 the battery give off more energy than it had previously accumulated (curve 1).

The velocity curves for two acceleration events are plotted in Fig. 10 and, for purposes of comparison, are shown with the curves for two medium class passenger cars (Opel Rekord 1900 and Daimler Benz 200/8D) and with a test point of a comparable electric car (BMW 1600 E).

The comparison with the traditional passenger cars shows that indeed the maximum acceleration of the hybrid passenger car is fairly good, however the total acceleration time is rather large because of the low acceleration at the start and in the upper velocity range. Here too, for example, a third gear would

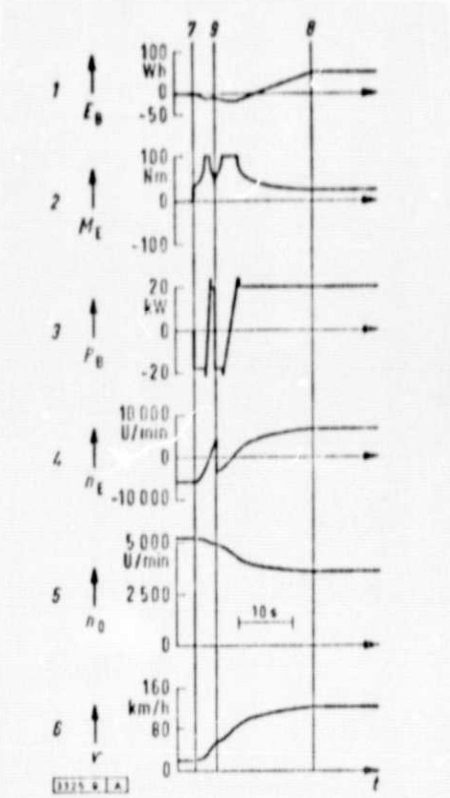

Fig. 9. Acceleration from 20 to 124 km per hour with an initial engine speed of n_O = 5000 rpm (analog computer result). For data not shown here see the key to Fig. 8.

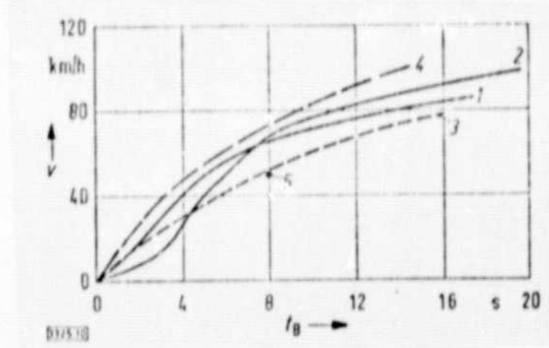

Fig. 10. Acceleration comparison for various cars.
1. Passenger car with hybrid propulsion system with n_O = 2500 rpm
2. Passenger car with hybrid propulsion system with n_O = 3350 rpm
3. Daimler Benz 200 D
4. Opel Rekord 1900
5. Test point of a BMW 1600 E electric passenger car.

result in a considerable improvement.

The acceleration is already at this time clearly better than that of the BMW 1600 E electric passenger car.

To conclude this discussion of steady and intermittent motive outputs a few figures on the basic propulsion system outputs should be given. Ignoring the gyro drive losses and gear losses they are obtained from the nomogram (Fig. 6) or by means of equation (6) and are tabulated in Table 1 as a function of the initial speed of the i.c. engine n_{Oideal}.

The share of the maximum acceleration output of 125 kW at n_{Oideal} = 5000 rpm contributed by the gyro drive at the beginning of the acceleration process is about 100 kW, since the output of the i.c. engine is regualted (stabilized) very slowly for emission reasons. The maximum specific output of the gyro drive is thus about 2 kW/kg. It is one multiple higher than that of the i.c. engine (0.7 kW/kg) and that of the electric motor (0.3 kW/kg without direct current regulator and battery). In spite of the rather heavy electro components this high specific output of the gyro drive results in a completely satisfactory weight per output of the complete hybrid propulsion system.

5. Energy Balance During Intermittent Driving Conditions

Definite and reproducible driving cycles are suitable for energy considerations. The following tests are based on the Europe cycle.

In Fig. 11 the velocity v corresponding to the Europe cycle for the car in question with a hybrid propulsion system (1300 kg) is plotted (curve 6). The speed n_0 of the i.c. engine (curve 5) is n_{0ideal} = 2500 rpm at the start, changes during the cycle between the limit values of 2630 and 1500 rpm, and at the end of the cycle has again reached the initial engine speed. The output P_0 of the i.c. engine (curve 4 fluctuates between 0.5 and 3 kW, and that of the battery (curve 3) between P_B = 10 and -12 kW. For the ideal engine speed n_{0ideal} selected here, the i.c. engine supplies an amount of energy during the cycle equal to E_0 = 7500 Wh (curve 2) and the battery supplies E_B = 5 Wh (curve 1).

Thus for driving once through the cycle the energy expended is $E'_{cycle} = E_0 + E_B$ = 80 Wh. This takes into account all output losses with the exception of the mechanical losses in the gears. With these the actual energy expended comes out

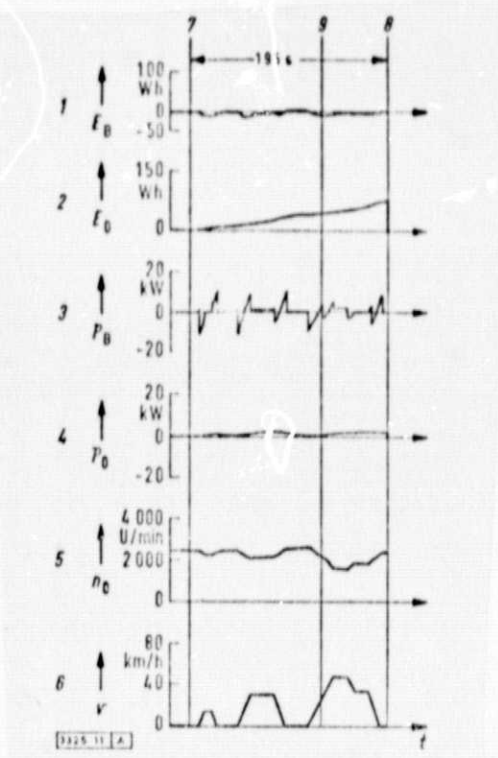

Fig. 11. Output and energy exchange when driving through the Europe cycle (analog computer result)

1. Energy consumption (battery) $E_B(t)$
2. Energy consumption (i.c. engine) $E_0(t)$
3. Output (battery) $P_B(t)$
4. Output (i.c. engine) $P_0(t)$
5. Speed (i.c. engine) $n_0(t)$
6. Velocity v(t)
7. Start of cycle
8. End of cycle
9. Shift point

to:

$$E_{cycle} = 91 \text{ Wh}.$$

On the other hand, for this cycle and for this vehicle, taking into consideration the mechanical gear losses, we get the following energy breakdown:

for accelerations	$E_a = 58$ Wh
for air and wheel resistance	$E_L + E_R = 64$ Wh
total energy to be spent without recovery	122 Wh.

The energy expenditure is thus reduced due to recovery (regenerative braking) by an amount equal to 122 Wh-91 Wh = 31 Wh or 25.4% (the comparable figure for the electric car is about 10%). With respect to the acceleration energy which can maximally be recovered, this gives a recovery efficiency of $\eta_{rec.} = 0.71$.

The fact that with this hybrid propulsion system the predominant portion of the acceleration energy without conversion into other forms of energy is exchanged mechanically between the vehicle and the gyro drive has, as expected, a very favorable effect on the recovery efficiency.

The ratio of the portion of the energy $\upsilon_e = E_E/E_O$ to the total cycle energy (80 Wh) is υ_e = 5 Wh/75 Wh = 0.067 in the example in Fig. 11. For any given propulsion system, by appropriately selecting the gear ratios i the speed of the i.c. engine n_{Oideal} and the shift speed can be controlled approximately within the following limits: /11

$$0 \leq \upsilon_e \leq 0{,}6.$$

The case of $\upsilon_e = 0$ means that energy -- as for a vehicle with an internal combustion engine -- can only be stored in the form of fuel, whereas in the case of $\upsilon_e = 0.6$ 40% of the energy can thus be stored in the form of electricity by recharging the battery.

Table 1. Comparison of outputs for a hybrid passenger car.

Type of output	Unit	Initial engine speed n_{Oideal} in rpm		
		2500	3350	5000
maximum output for acceleration (short period peak output) $P_{max}=P_O+P_E+P_G$	kW	74	92	125
medium term output of the i.c. engine and electric motor (until battery is discharged) $P_{med} = P_O+P_E$	kW	21	25	31
long term output (until tank is empty) $P_{long} = P_O$	kW	10	14	20

Table 2. Energy comsumption for extrapolated Europe cycle.

Operation with	Fuel consumption 1/100 km	electric energy consumption kWh/100 km
$\upsilon_e = 0$	6.5	--
$\upsilon_e = 0.8$	5.2	1.9
$\upsilon_e = 0.6$	3.9	3.8

Extrapolating the Europe cycle ($v_{average}$ = 18 km per hour, s = 975 m) to a driving distance of 100 km we get the following energy consumption levels with a partial throttle consumption of the i.c. engine of 450 g/kWh (these values are listed in Table 2):

In the case where υ_e = 0, with 6.51 1/100 km, we thus get a fuel consumption of about 50-60% of the consumption of a comparable passenger car with a traditional propulsion system (fuel consumption in stop-and-go operation about 11-13 1/100 km).

When the hybrid car is using only electric energy (υ_e = 1) the gyro effect cannot be used. Therefore this mode of operation is sensible only in special cases.

To be sure, an alternative to the electric car arises when the i.c. engine of the G.E.I. hybrid propulsion system discussed here is replaced by a second electric motor (G.E.E. hybrib propulsion system). Analysis of such a hybrid propulsion system on an analog computer resulted in a longer range than that for a comparable electric car because of the high recovery efficiency and the reduced peak load of the battery with better motive outputs. Since the second, larger electric motor needs to be operated only with field control, i.e. the direct current regulator is to be regulated only for the smaller output of the first electric motor, such a G.E.E. hybrid version may also be more advantageous, with respect to production costs, than a car with just an electric propulsion system.

6. Completed Experimental Propulsion System

Based on the positive results of these theoretical studies a G.E.I. hybrid propulsion system was constructed and, to begin with, installed in a few test cars for bench testing and actual driving tests. The arrangement of the components was chosen in such a way so that the compact propulsion block was obtained which can be subsequently installed in a few assembly-line produced passenger cars and buses in place of the internal combustion engine. Fig. 12 shows a picture of the propulsion[1]. The gyro /12 component sits on top of the differential gear.

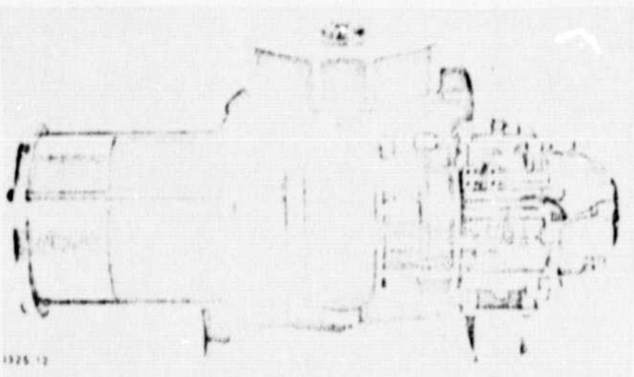

Fig. 12. Propulsion system, completed test version.

The mass of the propulsion system is 240 kg. This high weight is due to the fact that in the construction and design of this experimental propulsion system and its components considerations such as flexibility with respect to gear ratios and adaptability to various types of components and vehicles had priority. In a final version taking into consideration the weight factor (e.g. constructing the propulsion casing out of light metal instead of cast iron) the weight of the propulsion system consisting of the electric motor, i.c. engine, gyro component, differential and shift gears, will be reduced to about 160 kg. This value corresponds to that of a

1. The output of the Wankel engine installed here is only 15 kW.

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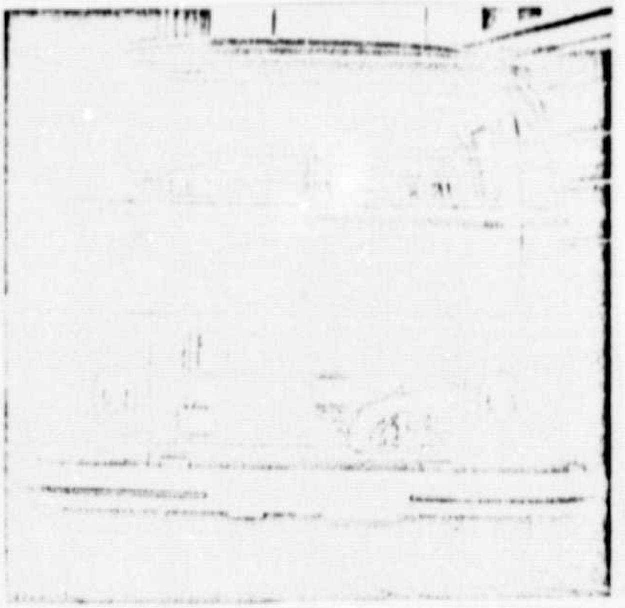

Fig. 13. Hybrid propulsion system in the experimental vehicle.

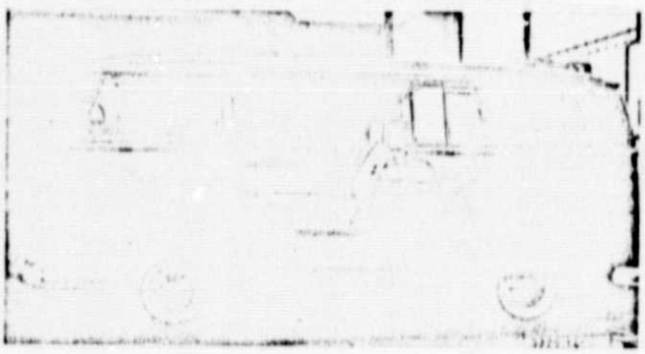

Fig. 14. Battery for the hybrid propulsion system in the experimental vehicle.

comparable internal combustion engine with all accessory apparatus for a passenger car.

In this weight comparison, power electronics and battery were not yet taken into account. As a basis for evaluation it can be assumed that the weight of the power electronics is compensated for by reductions in the weight of the vehicle (e.g. by reducing the size of the tank and radiator). However, the weight of the battery (i.e. 150 kg in the case of the lead battery selected here) in this hybrid propulsion system is added to the empty weight of the vehicle and thus reduces the disposable load. for example from 450 to 300 kg.

The conservative assumption of an empty weight of 1200 kg discussed in section 3.1 thus turns out to be completely realizable -- the empty weight of the Audi 100 is 1050 kg. An example of this experimental propulsion system was first of all installed in a VW bus in the context of a demand-bus project [14]. Fig. 13 shows the installed propulsion system and fig. 14 shows the installation of the accompanying battery underneath the middle seat.

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