TESTING OF ELEVATOR MACHINES

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ABSTRACT
The paper deals with the principal aspects of the testing of geared elevator machines in laboratory conditions. A brief description and illustration of the testing equipment is included as well as a summary of quantities measured during the tests. The machine load capacity was determined in respect of the temperatures of several reference places. The graphs were plotted from the measured values and compared with the theoretical exponential temperature/time diagrams. The measurements were taken while the machine worked in various conditions (variable braking torque and operation mode). The overall efficiency of elevator machine has also been determined.

1 INTRODUCTION. THE TECHNICAL PARAMETERS OF THE TESTED MACHINE

The principal aspects of the testing of geared elevator machines are dealt with in general and the methods and results of tests regarding a new elevator machine in particular. This machine was subjected to short-term load tests in a laboratory of the Technical University of Prague recently in order to determine its load capacity and mechanical efficiency.

The machine will be used for passenger elevators of the capacity of 320 kg and the rated speed of 0,7 m/s. It is equipped with a squirrel cage induction motor of the output of 3.5 kW and speed of 1500 r.p.m. (synchronous). The gear ratio of the worm gearing is 42 and the pitch diameter of the traction sheave 400 mm. The machine is of modern compact design. The high speed shaft is supported by two bearings with the rotor mounted on one end and the brake drum on the opposite end of the shaft. The stator is flange-mounted. The brake is of a conventional design, equipped with two brake shoes.

2 THE TESTING EQUIPMENT AND A SURVEY OF QUANTITIES MEASURED DURING THE TESTS

The tests were taken on a testing stand, the diagram of which is illustrated in Fig.1 together with the indication of some of the measured quantities. The power transmission from the elevator machine to the stand was accomplished by chain drive with the ratio of 1:1. The drive sprocket was secured by bolts to the traction sheave of the machine, while the driven sprocket was mounted on the upper shaft of the stand, around which the rocking part of the stand may swing. The power is transmitted to a direct current generator, used as a braking dynamo, via a planetary gearbox and vee-belt drive. The braking torque on the low speed shaft was induced by the rheostatic braking of the generator and sensed by tensometric sensing units in either direction.

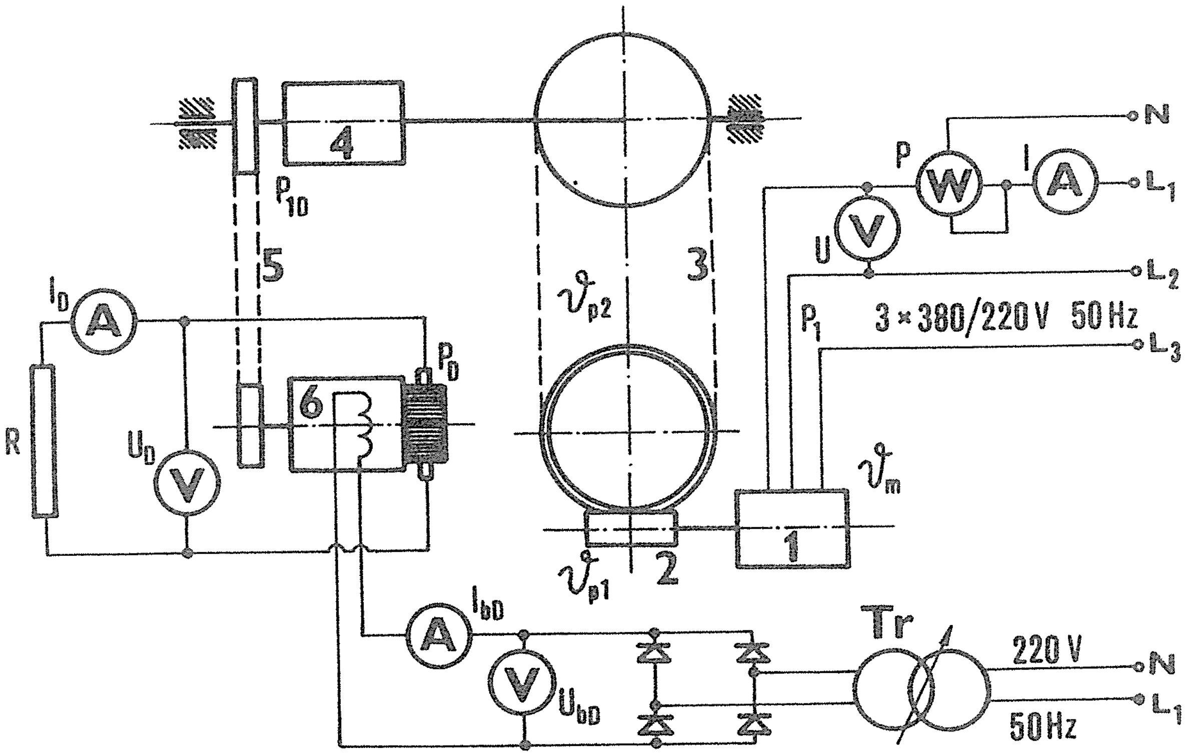

FIGURE 1: Diagram of the testing stand

The designation of individual components of the machine and testing equipment in Fig.1 is as follows:

1 . . . elevator motor
2 . . . gear reducer
3 . . . chain drive
4 . . . planetary gearbox (epicyclic gearing)
5 . . . vee-belt drive
6 . . . direct current generator (braking dynamo)

On the steel structure of the stand a special device for inducing a vertical force on the sheave is mounted to simulate the actual vertical load on sheave in elevator operation.

During the tests the following electrical and mechanical quantities were measured:
frequency and voltage of the mains
current and voltage of the generator, current and voltage of the exciting circuit

power input, current and speed (r.p.m.) of the motor
current and voltage of the brake magnet
braking torque

Furthermore temperatures of several reference places were observed, namely in the area of the tooth engagement of the worm gearing, of oil at the bottom of the gearbox and of stator winding in addition to the ambient temperature.

3 THE PRINCIPLE OF LOAD TESTS

The machine load capacity was determined in respect of the temperatures of the reference places mentioned above.

The temperature/time relation is in general of exponential characteristics, the analytic formulation of which leads to the equation

$$\vartheta = \vartheta_\infty \times \left(1 - e^{-\frac{t}{\tau}}\right) + \vartheta_0 \times e^{-\frac{t}{\tau}} \qquad (^\circ C),$$

where ϑ is temperature in a reference place in time t ($^\circ$C)
ϑ_∞ is temperature in a reference place in time approaching infinity ($^\circ$C)
ϑ_0 is ambient temperature ($^\circ$C)
τ is time heating constant (s)

$$\tau = \frac{m \times c}{\alpha \times S} \; (s),$$

where m is mass of the component, the temperature of which is observed (kg)
c is thermal capacity ($J \times kg^{-1} \times {}^\circ C^{-1}$)
α is coefficient of heat transfer ($J \times m^{-2} \times {}^\circ C^{-1} \times s^{-1}$)
S is surface area exposed to the cooling effect (m^2)

As apparent from the temperature/time equation, the curve approaches a horizontal asymptote, the ordinate of which is equal to the temperature ϑ_∞. If we consider the temperature increments only, i.e. $\Delta\vartheta$, assuming $\vartheta_0 = 0$, then we can easily calculate the magnitudes of temperature increments $\Delta\vartheta$ for $t = \tau$ and multiples of the time heating constant τ.

For $t = \tau$ $\quad \Delta\vartheta = 0,632\ \vartheta_\infty$
$t = 2\tau$ $\quad \Delta\vartheta = 0,864\ \vartheta_\infty$
$t = 3\tau$ $\quad \Delta\vartheta = 0,95\ \vartheta_\infty$
$t = 4\tau$ $\quad \Delta\vartheta = 0,98\ \vartheta_\infty$

The results are illustrated in Fig.2.

In the range of temperatures achieved during the tests to attain the quadruple of the constant τ would mean also to attain a steady state of the heat transfer. The state of the heat transfer is usually considered steady when the temperature gradient does not exceed 2 $^\circ$C per hour. The determination of the time heating constant τ is very difficult and not reliable unless it is carried out experimentally.

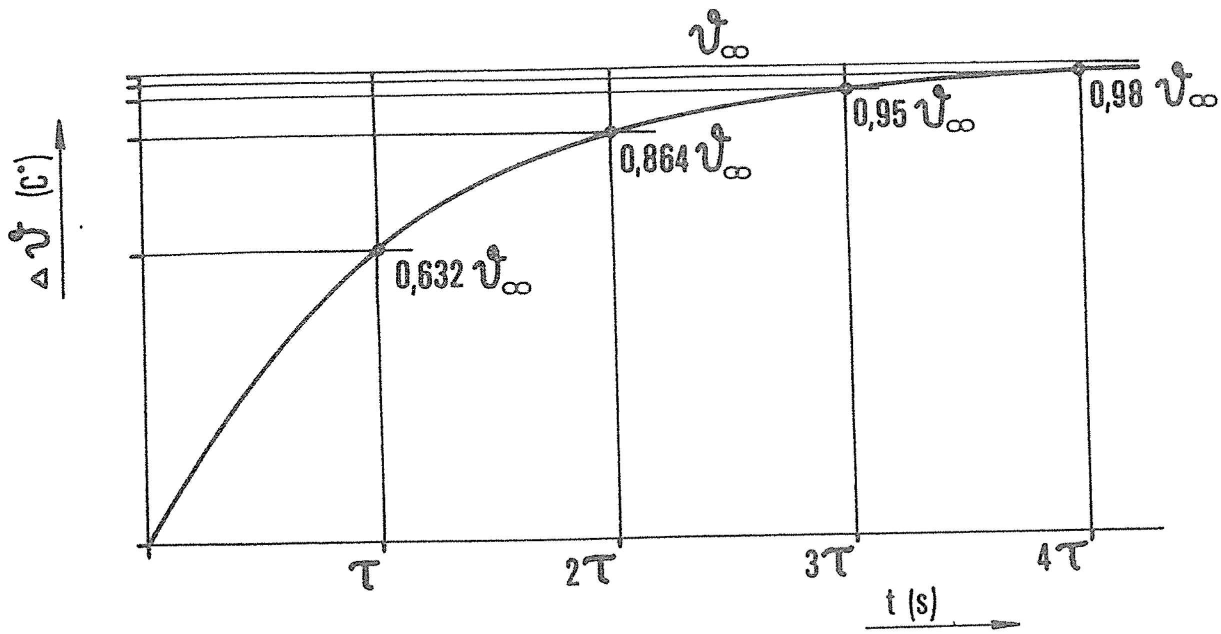

FIGURE 2: Temperature increments/time diagram

During the load tests the steady state was not achieved in all cases, however, it was possible to estimate the magnitudes of ϑ_∞ with satisfactory accuracy. The maximum temperature ϑ_∞ is in a definite place under definite operation conditions constant regardless of the initial temperature. If the test commences at higher temperature, the steady state is achieved earlier. This fact was advantageously utilized during the load tests.

4 THE PROGRAMME AND RESULTS OF LOAD TESTS

The tests were taken for three operation modes, represented by two principal parameters: load factor ε and the number of starts z.

The selected combinations are summarized:

i) ε = 60 % , z = 120 per hour

Schedule: 18 seconds of operation + 12 seconds of standstill, always two trips in one direction followed by two trips in reversed direction

ii) ε = 60 % , z = 60 per hour

iii) ε = 40 % , z = 60 per hour

With each operation mode the tests were taken for 7 values of the braking torque, quoted in Tab.1.

TABLE 1: Values of the braking torque (Nm)

M_1	164.8
M_2	268.8
M_3	416.7
M_4	518.7
M_5	681.5
M_6	826.7
M_7	922.9

In Fig.3 a graph illustrates temperatures of reference places/ time dependence for variant i) and braking torques M_1 up to M_5.

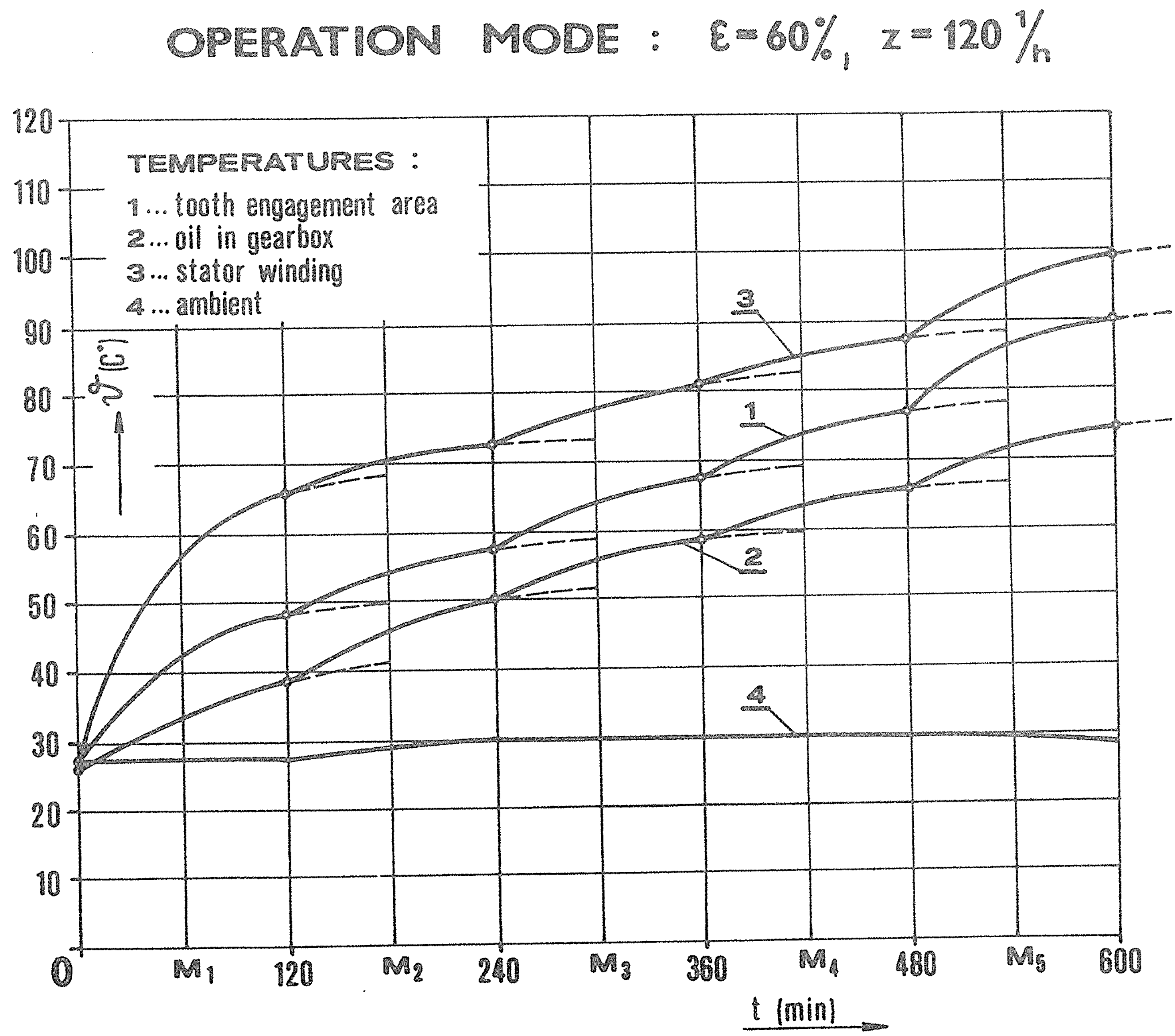

Fig.3: Variant i). Temperatures/time diagram for braking torques M_1 to M_5

The same tests for braking torques M_6 and M_7 followed later. The initial temperatures were higher to shorten the testing time. Temperature/time diagram is depicted in Fig.4.

As seen from Fig.3 and Fig.4 the theoretical exponential dependence between both quantities was fully confirmed by practical tests. The last test under the maximum load (braking torque M_7, actually representing overloading of the machine), must have been interrupted after it had run for half-an-hour only, as the temperatures in all reference places continued to rise into dangerous zone and the temperature of oil in the gearbox exceeded the maximum permitted value (ambient temperature + 55°C, as specified by the oil supplier).

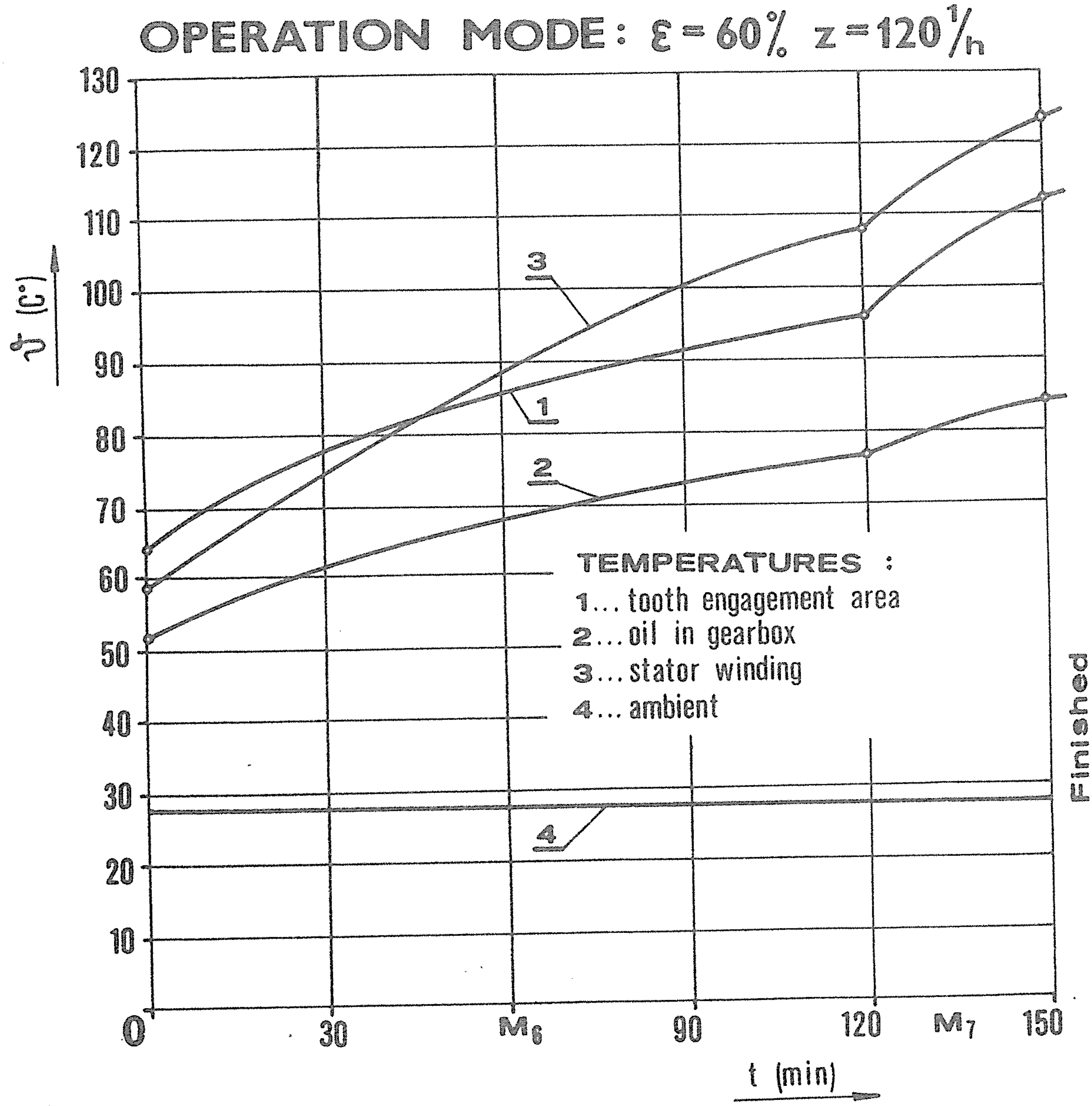

Fig.4: Variant i). Temperatures/time diagram for braking torques M_6 and M_7

For completion a brief information about the results of load tests for variants ii) and iii) is also submitted here as a comparison with the results of variant i) may be found instructive. A graph in Fig.5 depicts the temperatures/time diagram for variant ii) and braking torques M_3 and M_4, while in Fig. 6 a similar graph for variant iii) and braking torques M_5 to M_7 is illustrated. With variant iii) temperatures for lower braking torques M_1 to M_4 were of little significance and not recorded in the graph.

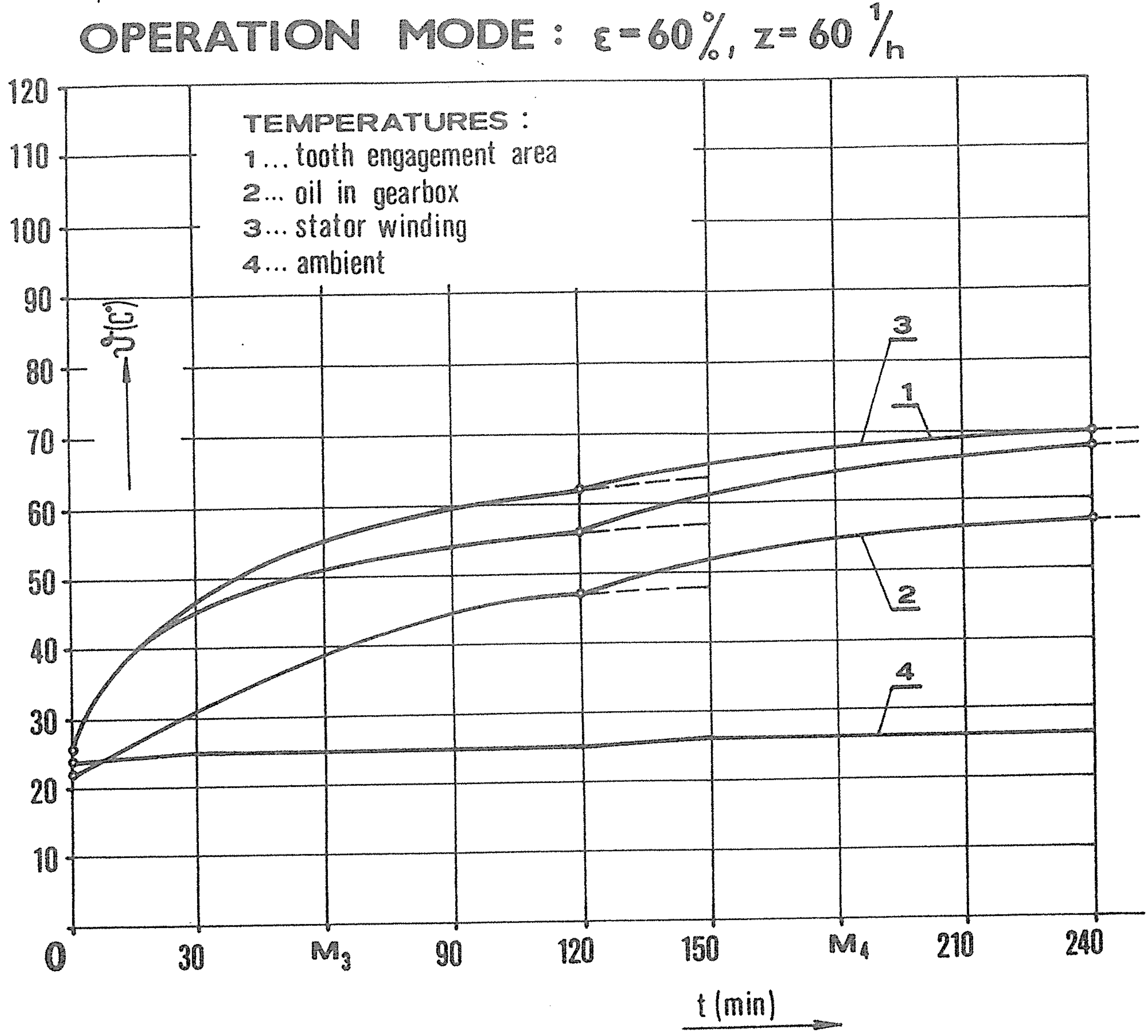

Fig.5: Variant ii). Temperatures/time diagram for braking torques M_3 and M_4

If a sufficient number of partial load tests is taken, graphs expressing the dependence of maximum temperatures ϑ_∞ (or maximum temperature increments $\vartheta_\infty - \vartheta_0$) of each reference place on the braking torque may be plotted from the measured values for each operation mode. From such a graph the maximum temperature increment for a definite reference place may be read off for any braking torque.

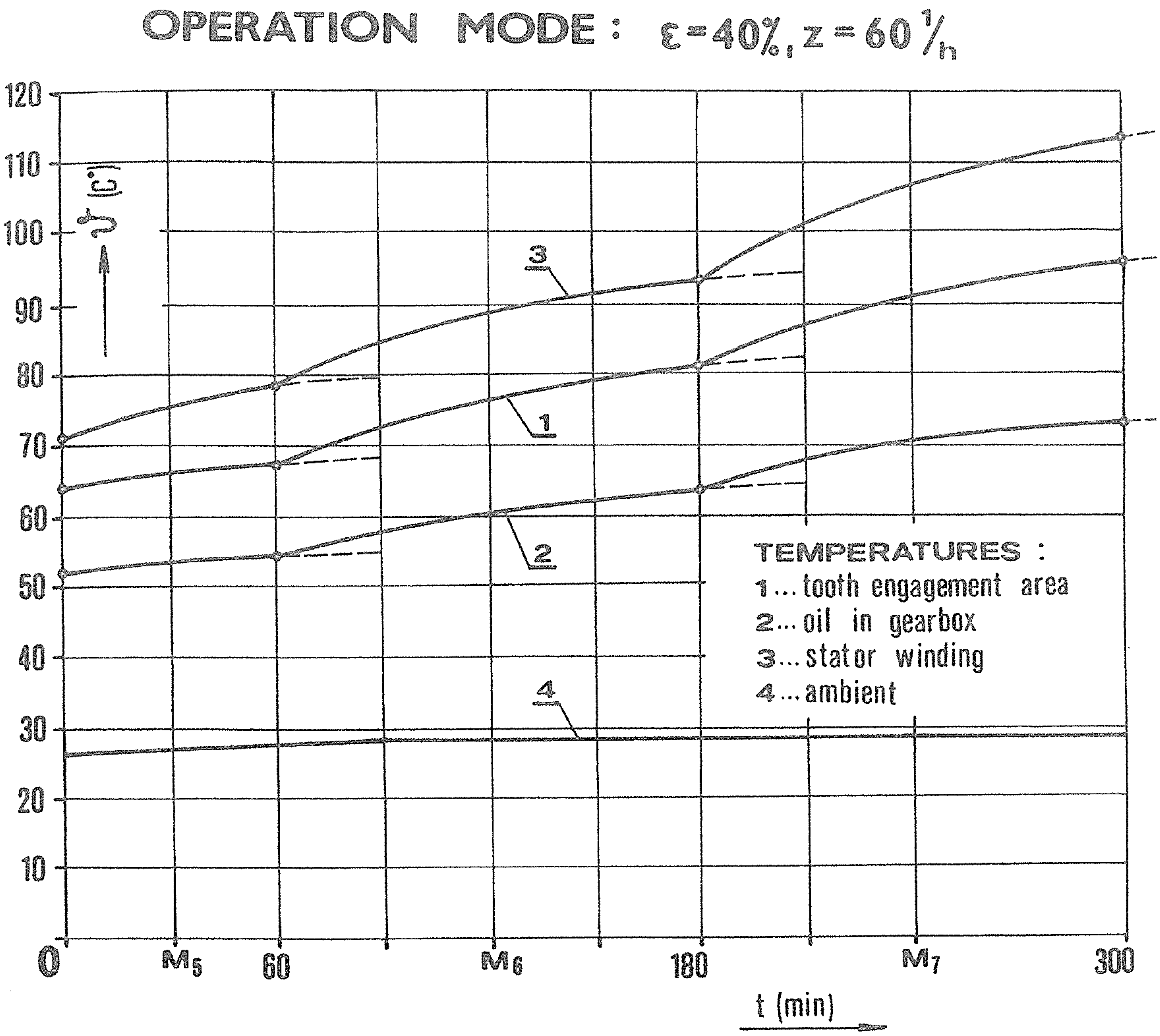

Fig.6: Variant iii). Temperatures/time diagram for braking torques M_5 to M_7

5 THE DETERMINATION OF THE EFFICIENCY OF THE MACHINE

The efficiency was determined by measurements of electrical and mechanical quantities and subsequent calculation several times during the tests, gradually for all 7 values of the braking torque.

In Tab.2 the values of the overall efficiency η of the elevator machine are tabulated in dependence on the braking torgue M_i and the efficiency/braking torque dependence is illustrated in Fig.7.

For the calculation a simple formula was used, namely

$$\eta = \frac{M_i \times n}{9550\, P \times i \times \eta_c} ,$$

where n are r.p.m. of the motor
P is input of the motor (kW)
i is gear ratio
η_c is mechanical efficiency of chain drive (estimated as 0.96)

TABLE 2: Efficiency of elevator machine η

Braking torque M_i (Nm)		η
M_1	164.8	0.4309
M_2	268.8	0.5280
M_3	416.7	0.5878
M_4	518.7	0.6054
M_5	681.5	0.6160
M_6	826.7	0.6088
M_7	922.9	0.5906

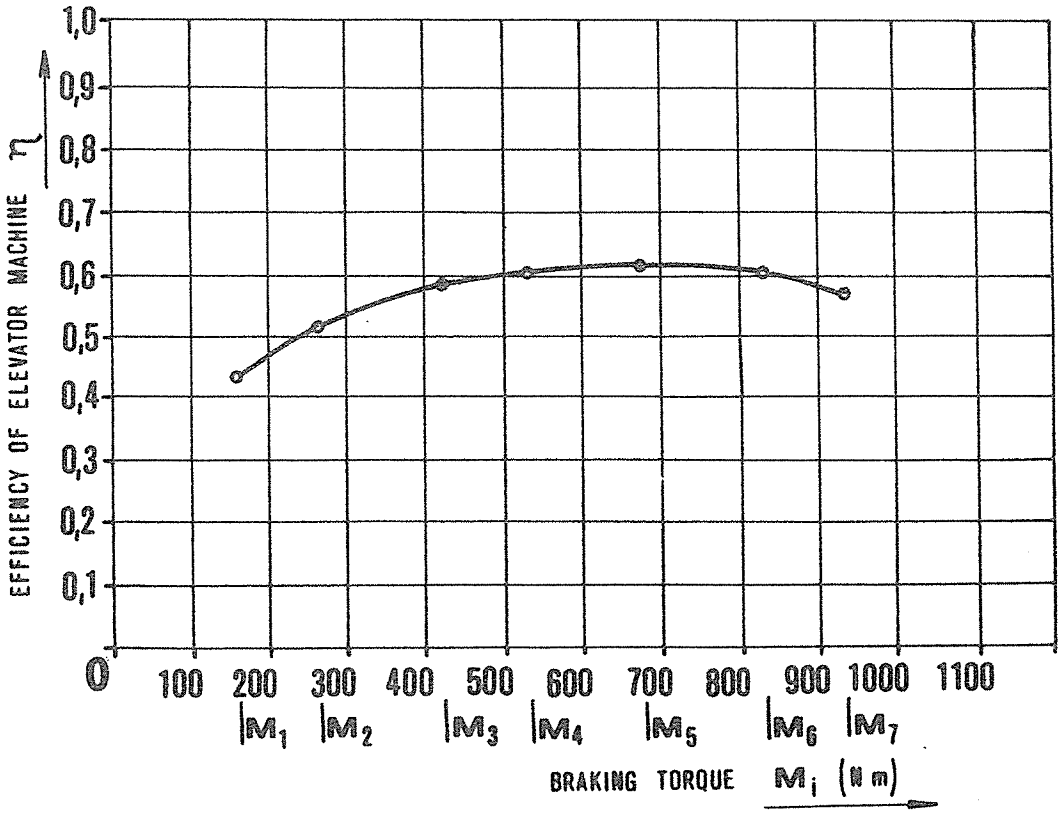

Fig.7: Overall machine efficiency/braking torque diagram

6 CONCLUSION

The load tests proved the elevator machine to be of optimum design. When the heaviest operation mode was applied during the tests, represented by the load factor of 60 % and the number of starts 120 per hour, temperatures in reference places achieved the maximum permissible values in the zone of rated load.

The overall efficiency of the machine in the zone of rated load was rather high in contrast to elevator machines of preceding design as the value was slightly in excess of 0.6.

7 FUTURE TRENDS

Although the results of the first load tests of geared elevator machines in the university laboratory are very interesting, they have not brought new experience of general significance as in fact they were limited to one machine only.

The testing of elevator machines is proposed to continue. Another two machines will be under tests in 1988 and it is suggested to concentrate the attention to the following problems in addition to those discussed in this paper:

i) The investigation of the relationship between maximum temperature ϑ_{∞} and braking torque for each reference place.
ii) The distribution of the overall efficiency to individual components of the machine, e.g. the determination of the mechanical efficiency of the worm gearing. A new dynamometer installed in the laboratory will facilitate to take a separate test of the motor, so that partial efficiencies will be easy to determine.
iii) The dependence between the worm gear efficiency and the rubbing speed for the most common combinations of worm and worm wheel materials, surface finish, the kind of lubricant etc.
iv) The accomplishment of long-term tests in order to find out the relationship between
   a) the efficiency and the number of operations
   b) the noise level and the number of operations.

8 REFERENCES

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9 BIOGRAPHICAL NOTES

Lubomír Janovský has been Senior Lecturer at the Faculty of Mechanical Engineering, Technical University of Prague, since 1957. He is also an elevator and escalator consultant and has been a foreign correspondent for ELEVATOR WORLD since 1978. He has been Chairman or Member of various prototype, technical and standard draft committees. Is a Member of the Czech Technical Council for Elevators and a Member of the Steering Committee of the International Convention for Elevator Engineering.

He gained his Ing. (Masters level) in Mechanical Engineering in 1957 and his CSc. (doctorate) in Vertical Transportation, both from the Technical University of Prague.

He has written numerous books and papers on vertical transportation and materials handling.