EFFICIENCY TEST ON THE LARGE PLANETARY GEAR OF A COAXIAL BALL MILL DRIVE

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1. Introduction

Assessment of the drive system of a large ball mill from the point of view of energy economy leads to the coaxial drive layout. Dependability and maintenance costs are further decisive criteria. The high standards of engineering attained in gear drive construction can only be fully exploited in the self-contained coaxial gear unit.

The driving of important manufacturing plants and ships, land vehicles and aircraft is nowadays effected by use of self-contained gear units and high quality gearing. Ef ciency is the key requirement. Similar skeleton conditions are encountered in the drive systems of large ball mills. Hence the same assessment criteria are justified for ball mill drives.

2. Running costs and driving efficiency

Higher investment costs can usually be amortized within a short time by the lower running costs.

In cement manufacture an eye must be kept on the total energy requirements on the one hand, and on the converted energy in kWh on the other. As a rule, about 60% of the total electrical energy requirements are expended for the driving of the ball mills.

The significance of avoiding drive energy losses can be easily illustrated by the example of a production line of 1 million tons per year output. The total driving power for ball mills ... in the order of 10 MW.

At production rate of 8000 hours per year, poorer efficiency calls for more kilowatt-hours, viz.:

- 2.5%	2.0 million kWh
- 3.5%	2.8 million kWh
- 5 %	4.0 million kWh

With the extra energy saved, the costs of a considerably higher investment can be covered, or an accordingly greater amount of material milled.

The harnessing of electrical energy and the costs entailed will be assessed from various aspects, according to the nature of the supply.

With the various possible drive systems, the overall efficiency can in fact deviate as much as 5% from the optimum.

3. Efficiency of the complete drive system

Since the pertinent literature has dealt diversely with the efficiencies of various drive layouts and individual drive elements, it would seem of interest to undertake numerical full-load tests on a planetary gear of a large ball mill, and to determine the efficiency of the drive system, including the motor losses.

A suitable object presented itself with the drive unit of a ball mill in the Schwelgern cement works of the Rheinischen Kalksteinwerke GmbH, Wülfrath in Duisburg-Hamborn.

Transmitted power	5500 kW
Speeds	990/14.5 rpm
Mill	4.8 m diameter
	17.6 m long
Weight of steel balls	380 tons

Motor Gear Mill

$$E_2 = E_1 - (V_M + V_G)$$

$$\eta_A = \frac{E_2}{E_2 + (V_M + V_G)}$$

$$\eta_A = \eta_M \cdot \eta_G$$

where:

E_1	Power supplied to the electric motor
E_2	Power transmitted to mill neck
E_M	Power transmitted from motor to gear
V_m	Power losses in electric motor
V_G	Power losses in planetary gear
η_A	Overall efficiency
η_M	Motor efficiency
η_G	Gear efficiency

The only power available for overcoming the mill resistance torque is that left (= E_2) after deduction of the losses in the motor and the gear unit.

4. Motor losses

The losses in the electric motor are determined by the motor manufacturer via the usual practices in his own works.

For the 5.5 MW, 5.5 kV induction motor employed, the motor ma facturer stated the following efficiencies:

- at 4/4 load 97.1%
- at 3/4 load 97.1%
- at 1/2 load 96.9%

5. easuring method for the gear losses

For these measurements, which are stringently demanding, the method employed is already decisive for the reliability of the results. Fundamental considerations are therefore essential to the defining of the process.

Convection

Enveloping surface

Radiation

Conduction

Mill

Radiation

Oil outlet

Enveloping surface

Planetary Gear

Location lugs

Measuring of air temperature

Oil inlet

Motor

Fig. 2 Measuring principle. Heat dissipation losses via convection, radiation and conduction.

Energy losses in the gear result principally from the friction at the points of tooth mesh and in the bearings. Escape is by transformation into heat. Oil and housing temperatures rise, and with this the temperature gradient. Under thermally stable conditions the heat generated by the losses is equal to that escaping.

a)
$$\underline{V_G = V_{Oel} + V_{Umg}}$$

Where:

V_G gear losses

V_{Oel} Heat carried away by the oil

V_{Umg} Heat dissipated into the surroundings

The heat carried away by the coolant oil is given by:

b) $$V_{Oel} = Q_{Oel} \cdot \gamma_{Oel} \cdot c_{p_{Oel}} \cdot \Delta_{t_{Oel}}$$

where:

V_{Oel} Power loss carried away by coolant oil in kW

Q_{Oel} Coolant oil flow rate in dm^3/s

γ_{Oel} Specific weight of coolant oil at operating temperature in kd/dm^3

$c_{p_{Oel}}$ Specific heat of coolant oil at operating temperature in $kJ/kg/^oC$

$\Delta_{t_{Oel}}$ Oil temperature difference in ^oC

All the values contained in these formulae can be determined easily and accurately by measurement.

The heat dissipation to the surroundings is given by:

c) $$V_{Umg} = \frac{S \cdot \Delta_t \cdot A}{1000}$$

where:

V_{Umg} heat dissipated into surroundings in kW

S heat transfer coefficient in $W/m^2/^oC$

Δ_t temperature difference between housing surface and surrounding in ^oC

A heat transfer area in m^2

The hot housing dissipates heat into the surroundings via convection, radiation and conduction, as indicated in Fig. 2.

It ıs discovered that the greater portion of heat passes into the surroundings via convection, and a somewhat smaller portion by radiation. The gear location lug surfaces are relatively small, i.e. about 2% of the total housing surface. The heat passing directly to the foundation is therefore a negligible fraction of the total heat dissipated into the surroundings.

Heat transfer coefficients given in the pertinent literature cannot be applied to the object under investigation, as this would entail errors far in excess of tolerance. Uncertainty in measuring would increase the possible disagreements. Values employed should be basically such as are valid solely for this plant itself on the grounds of measurement.

These considerations led to the determining of the heat transfer coefficient for the total heat dissipation to the surroundings by way of measurement.

The gear efficiency investigation hence requires 2 measuring tests, viz.:

- on the test stand at no load, to determine the heat transfer coefficient S
- in the cement works, to determine full-load losses.

The gear tested is suitable for this method, as the governing conditions on the test stand and in the cement works are largely coincident. The heat transfer coefficient into the surroundings, determined by measurement, can be applied unconditionally for the full-load test.

6. No-load test on the gear test stand

The losses are measured directly and indirectly.

Dir :t measurement is via a calibrated torque dynamometer between the electric motor and the gear. A cross check is made by electrical measurement with the calibrated test motor. The result acquired under thermal equilibrium is equal to the total no-load losses.

For the indirect measurement, the temperature of the coolant oil at inlet and outlet are taken with precision mercury thermometers, and the oil flow rate via a calibrated flow rate gauge.

Calibrated pressure gauges are used for measuring the delivery pressure of the coolant oil pump.

The characteristic of the lubricating oil pump is measured on a special pump test stand.

Formula b) supplies a value for the heat carried away by the coolant oil.

Letting V_T be the total losses measured with the torque dynamometer, then:

d) $$\underline{V_T = V_{Oel} + V_{Umg}}$$

from which:

e) $$\underline{V_{Umg} = V_T - V_{Oel}}$$

From the power balance at thermal equilibrium, heat transfer coefficient S can be calculated viz.:

f) $$\underline{S = \frac{V_{Umg} \cdot 1000}{\Delta_t \cdot A}}$$

The emperature of the housing surface is measured at 28 characteristic points. Values between 30°C and 42°C were determined. Ambient temperature is measured in an enveloping surface around the housing, at about 1 metre distance from the same. (See Fig. 2). The readings are between 24°C and 30°C. Thermal equilibrium conditions were attained after a continuous running duration of 62 hours. Calculation of the heat transfer coefficient is made with due consideration of the temperature differences relevant to the individual surface parts.
In this manner, a heat transfer coefficient of $S = 11.9\ W/m^2/^oC$ was obtained.

It was furthermore determined:

- Total no-load losses $V_T = 37.0$ kW

Of these:

- carried away by coolant oil $V_{Oel} = 30.5$ kW
- dissipated into surroundings $V_{Umg} = 6.5$ kW

The energy relations at no-load are illustrated in Fig. 3.

Input power

Expelled power

Heat dissipated into surroundings via
K: convection St: radiation L: conduction

Fig. 3 Energy relations under no-load

7. Full-load test in cement works

In the full-load test, the heat carried away by the coolant oil is measured in the same way as on the test stand. The same measuring instruments are employed. The readings of interest are those which are taken at the same machine tem-

Under equilibrium conditions an oil inlet temperature of 42.5°C is determined. It is of the same order of elevation as that at no-load, where 43.2°C was measured.

Room temperature and housing surface temperature are measured with the same instruments and at the same points as in the no-load test.

The housing temperatures lie between 25°C and 40°C, the ambient air temperatures between 12°C and 20°C.

The attaining of thermal equilibrium conditions was both difficult and time consuming, as even the slightest disturbance in the coolant system and from the operating end required a long time to re-settle.

At rating output the losses carried away by the oil amount to:

$$V_{Oel} = 43{,}4 \text{ kW}$$

Those dissipated from the housing into the surroundings amount to:

$$V_{Umg} = 11.3 \text{ kE}$$

The total gear losses are hence

$$V_G = 54.7 \text{ kW}$$

The efficiency of the planetary gear alone is given by:

g)
$$\eta_G = \frac{(E_M - V_G) \cdot 100}{E_M}$$

$$\underline{\underline{\eta_G}} = \frac{(5515 - 54.7) \cdot 100}{5515} = \underline{\underline{99 \pm 0.1\%}}$$

From the pump characteristics the lubricating oil pump power is found to be:

$$P = 5.8 \text{ kW}$$

Th total losses, including the driving power of the lubricating oil pump, are given by:

$$V_{GP} = 60.5 \text{ kW}$$

The gear efficiency, allowing for the lubricating oil pump is given by:

h) $$\eta_{GP} = \frac{(E_M - V_{GP}) \cdot 100}{E_M}$$

$$\underline{\underline{\eta_{GP}}} = \frac{(5515 - 60.5) \cdot 100}{5515} = \underline{\underline{98.9 \pm 0.1\%}}$$

The efficiency of the whole drive unit (planetary gear and el tric motor) is given by:

i) $$\eta_A = \eta_{GP} \cdot \eta_M$$

$$\eta_A = (0.989 \cdot 0.971) \cdot 100$$

$$\eta_A = 96.0\%$$

Input power

Lost power

Heat dissipated into surroundings by
K: convection St: radiation L: conduction

Fig. 4: Energy relations under full load

From the test readings and the motor efficiency data, calcı ation and exterpolation supplies the relation between efficiency and load shown in Fig. 5. The cement manufacturer himself is interested in practically the full-load efficiency only.

Planetary gear without oil pump

Efficiency
in per cent

Planetary gear with oil pump

Induction motor

Complete drive system (motor and planetary gear)

Load

Fig. 5 Efficiency in relation to load
Coaxial drive of ball mill via MAAG planetary gear
and induction motor
5.5 MW; 990/14.5 rpm

8. Concluding remarks

The efficiency tests described in the aforegoing refer to the design features and the standard of manufacture of the MAAG planetary gear subjected to the tests. The results obtained cannot be applied to other coaxial drives deviating from this conception.

If, for example, for the same power the housing surface area is doubled and no oil return cooling is provided, the entire power loss is dissipated into the atmosphere and foundation. The 20% power loss from the housing to the surroundings in the case in hand would rise to 100%. The significance of the use of exactly determined heat transfer coefficients would also gain a corresponding amount.

Were the gear support surfaces relatively large, the loss by co[illegible]ction via the locating lugs would reach proportions which could no longer be neglected.

The viscosity of the oil has a substantial influence on the frictional losses. The tests were performed with the oil quality specified for operation at a relatively low operating temperature. A thinner oil or a higher operating temperature would result in lower losses and a higher efficiency. If, with the same oil, the inlet temperature were to rise 8°C, for example, the gear efficiency would increase about 0.1% from 99.0% to 99.1%.

The test results should provide the cement manufacturer with additional data for a sure assessment of the efficiency and energy situation for drive systems of large ball mills.

For the sake of completeness it should be stated that the MAAG planetary gears which have already been in service for years still have flawless teeth, so that the efficiency has suffered no deterioration since they were put into operation. That case-carburized teeth, ground with careful amounts of flank relief are not prone to wear in operation - likewise friction bearings - is already well known.

In contrast to the annulus drive layout, the inevitable mill displacements due to bearing wear, thermal deformation and sagging have no influence on the efficiency of the drive system. The induction motor power factor which must be allowed for in the efficiency calculation is far more favourable than with the annulus drive system.

Summary

The general level of the avoidable energy losses in present-day customary drive systems for large ball mills is shown by means of a numerical example. To determine the efficiency of the dual planetary gear, the energy lost in the form of heat carried away by the oil and dissipated into the surroundings is measured. The heat transfer coefficient between housing and surroundings is measured for the conditions applying directly to the gear tested.

The investigations supply the full-load efficiency of the planetary gear and of the whole drive unit comprising electric motor and gear.

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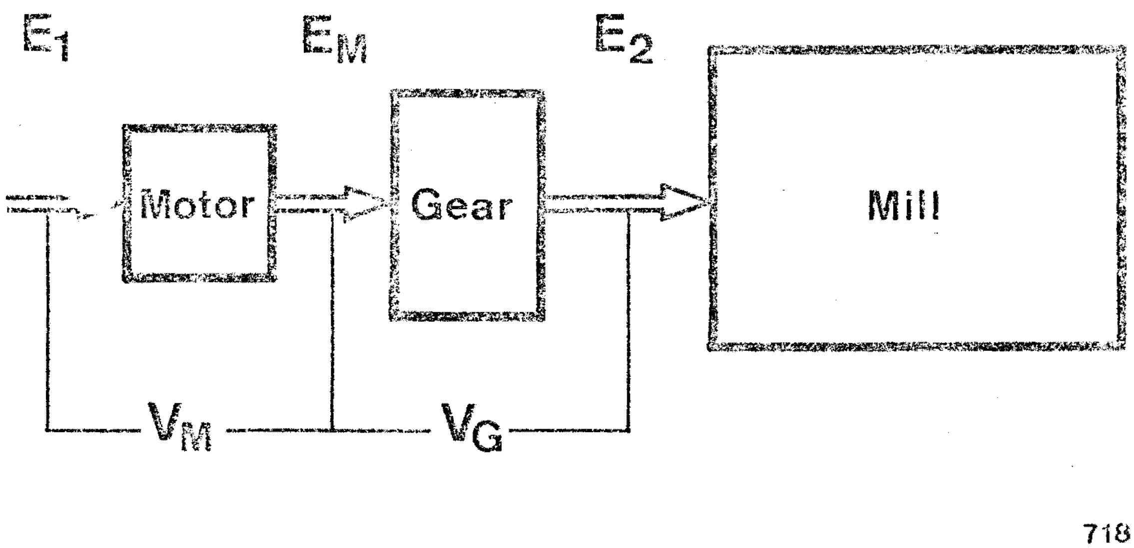


Fig. 1 Machine layout - Energy flow diagram

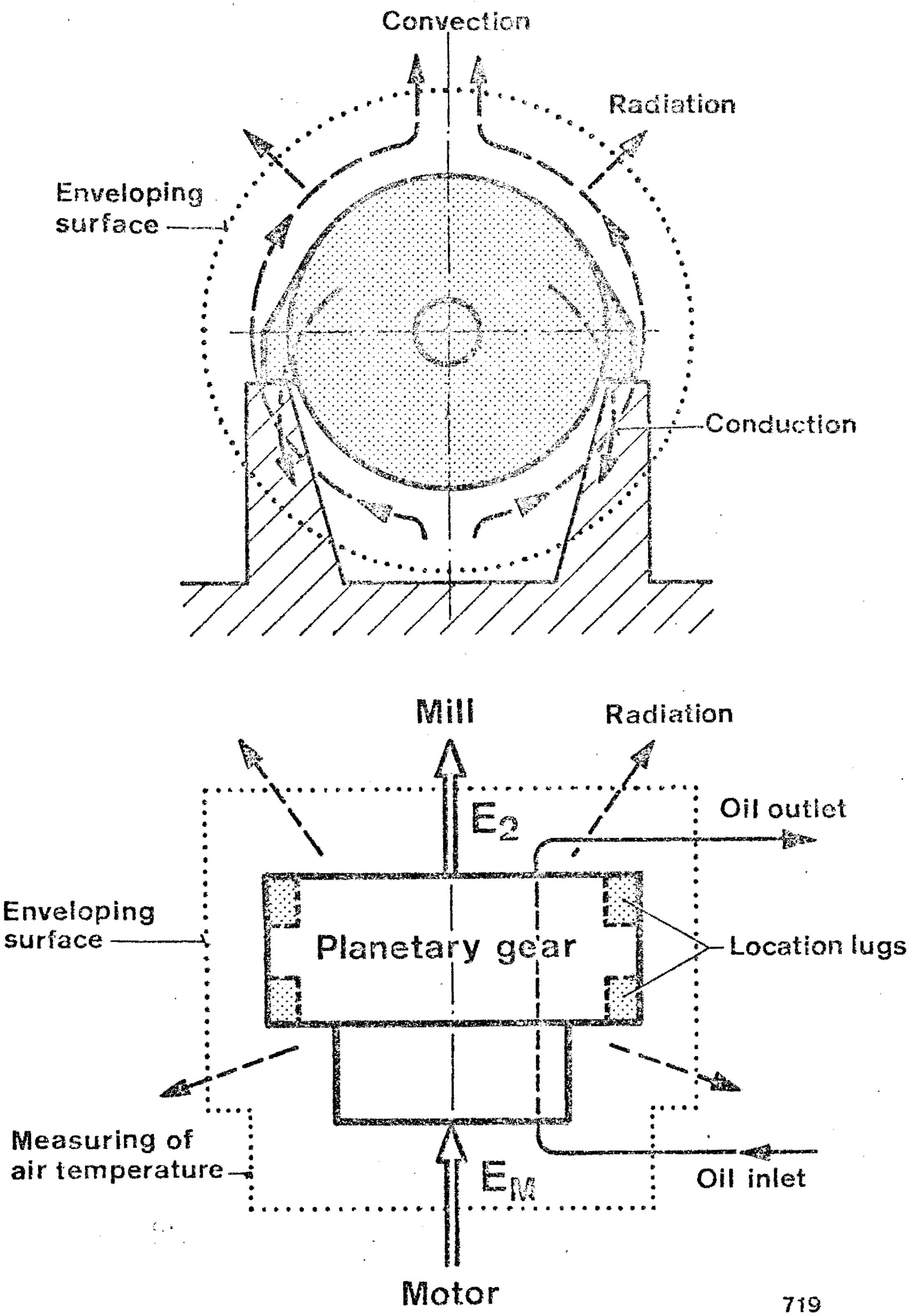


Fig. 2 Measuring principle. Heat dissipation losses via convection, radiation and conduction.

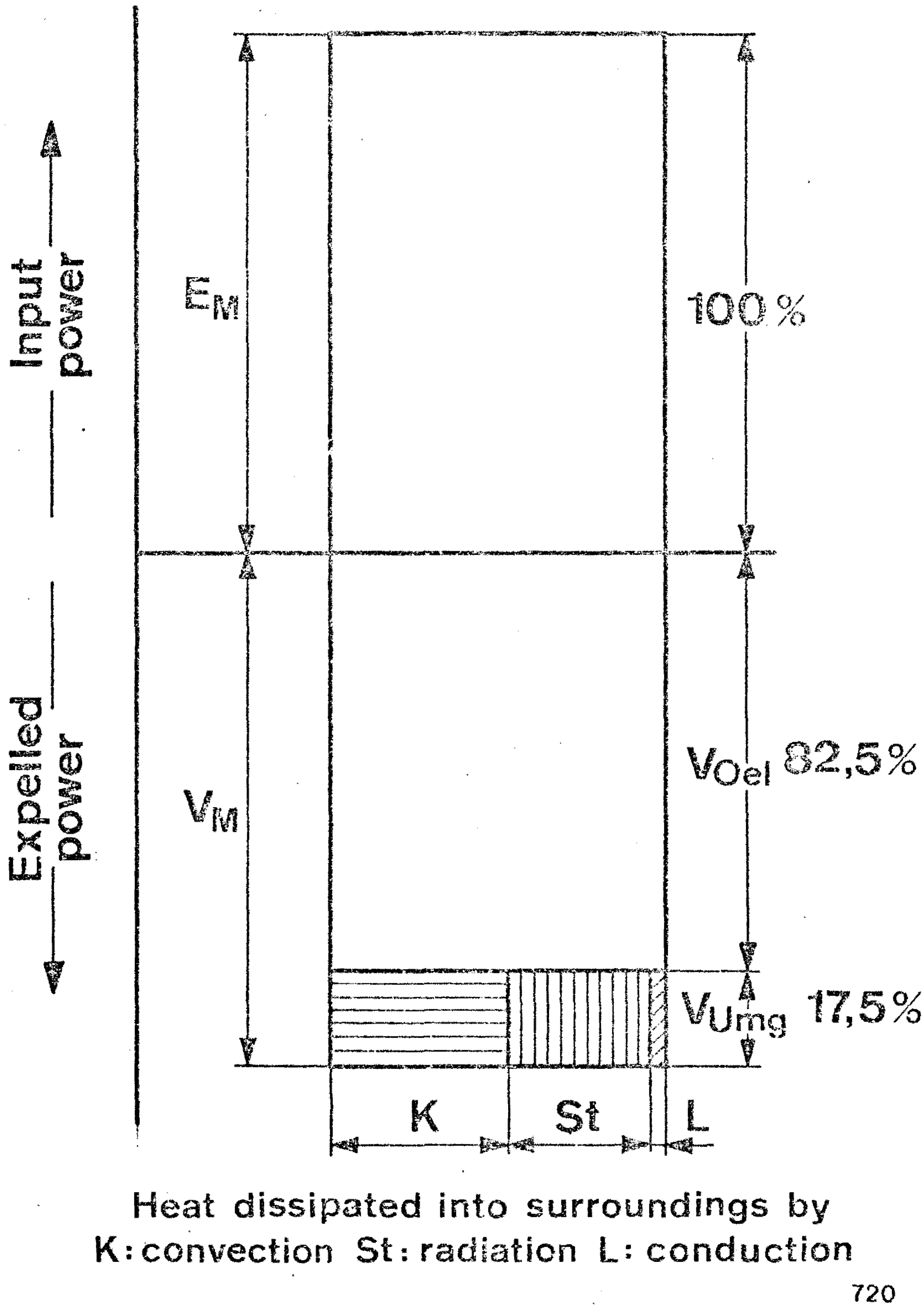
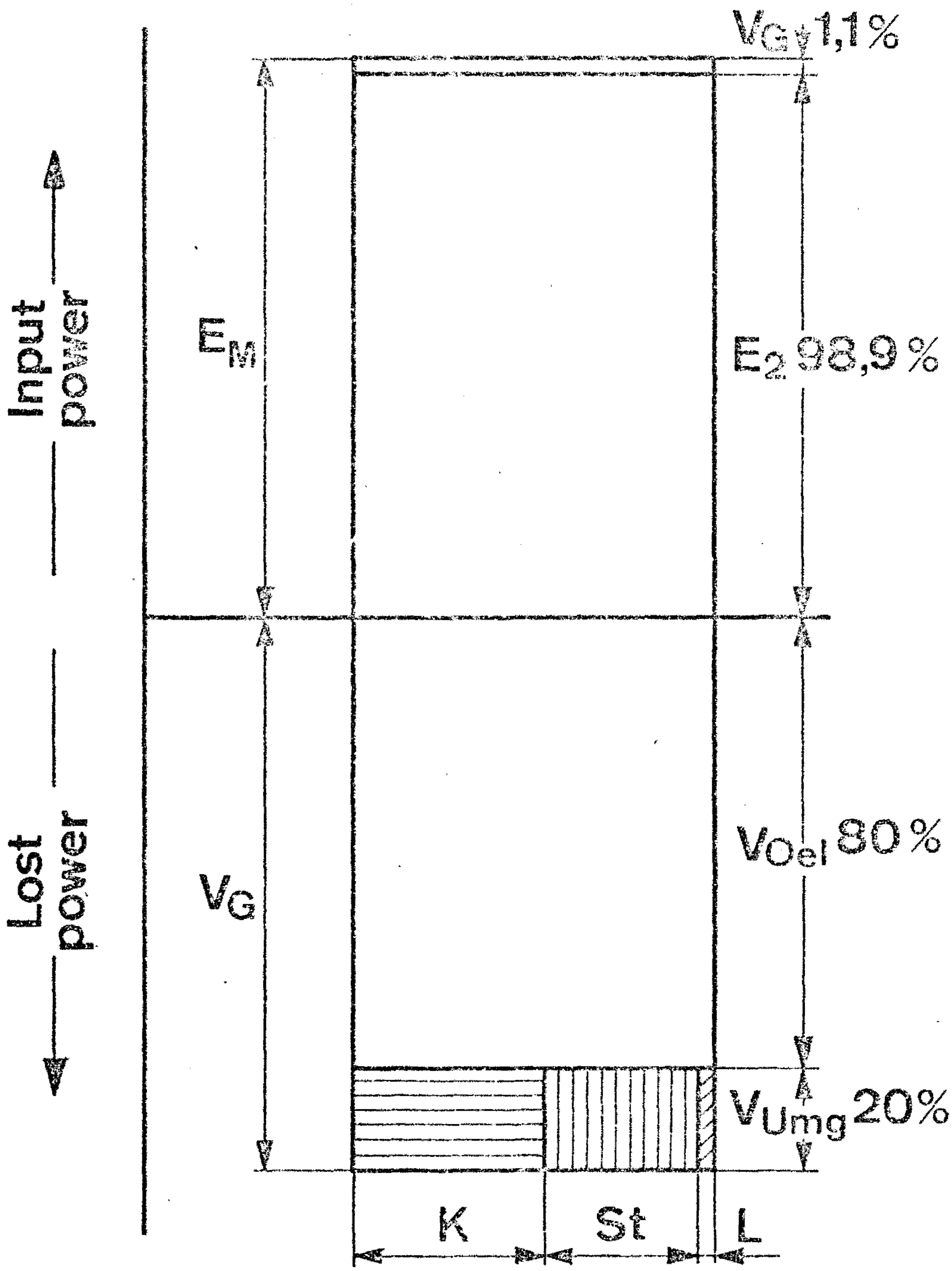


Fig. 3 Energy relations under no-load



Heat dissipated into surroundings by
K: convection St: radiation L: conduction

Fig. 4 Energy relations under full load

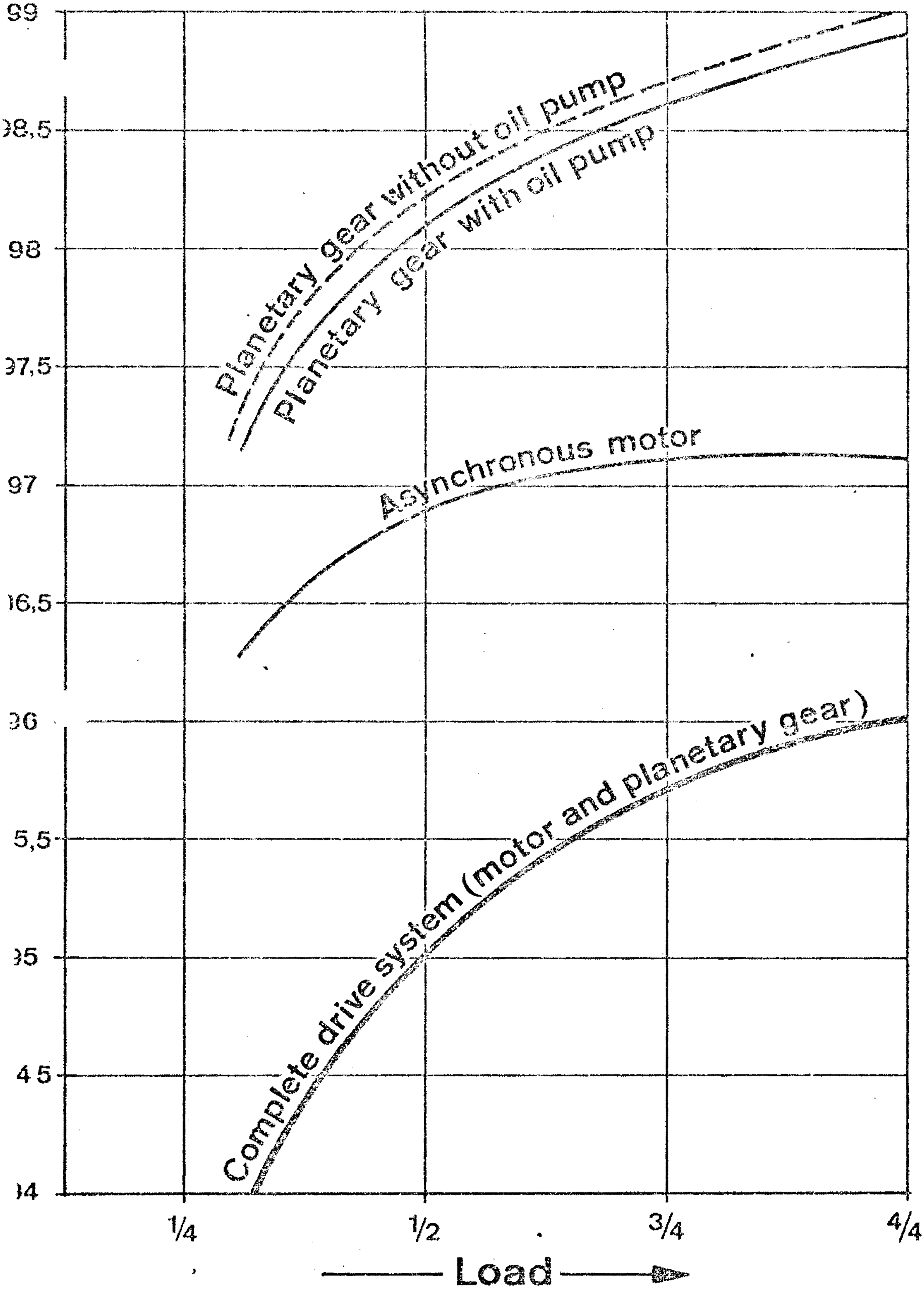


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Fig. 5 Efficiency in relation to load. Coaxial drive of ball mill via MAAG planetary gear and asynchronous motor 5,5 MW; 990/14.5 rpm.