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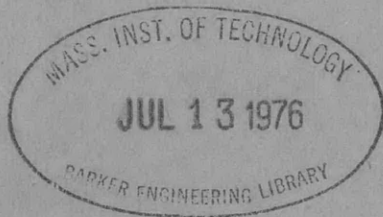
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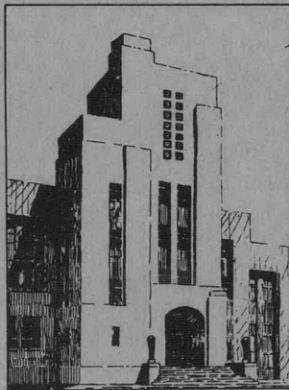
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LIGHTWEIGHT STEEL DESIGN OF MACHINE TOOLS



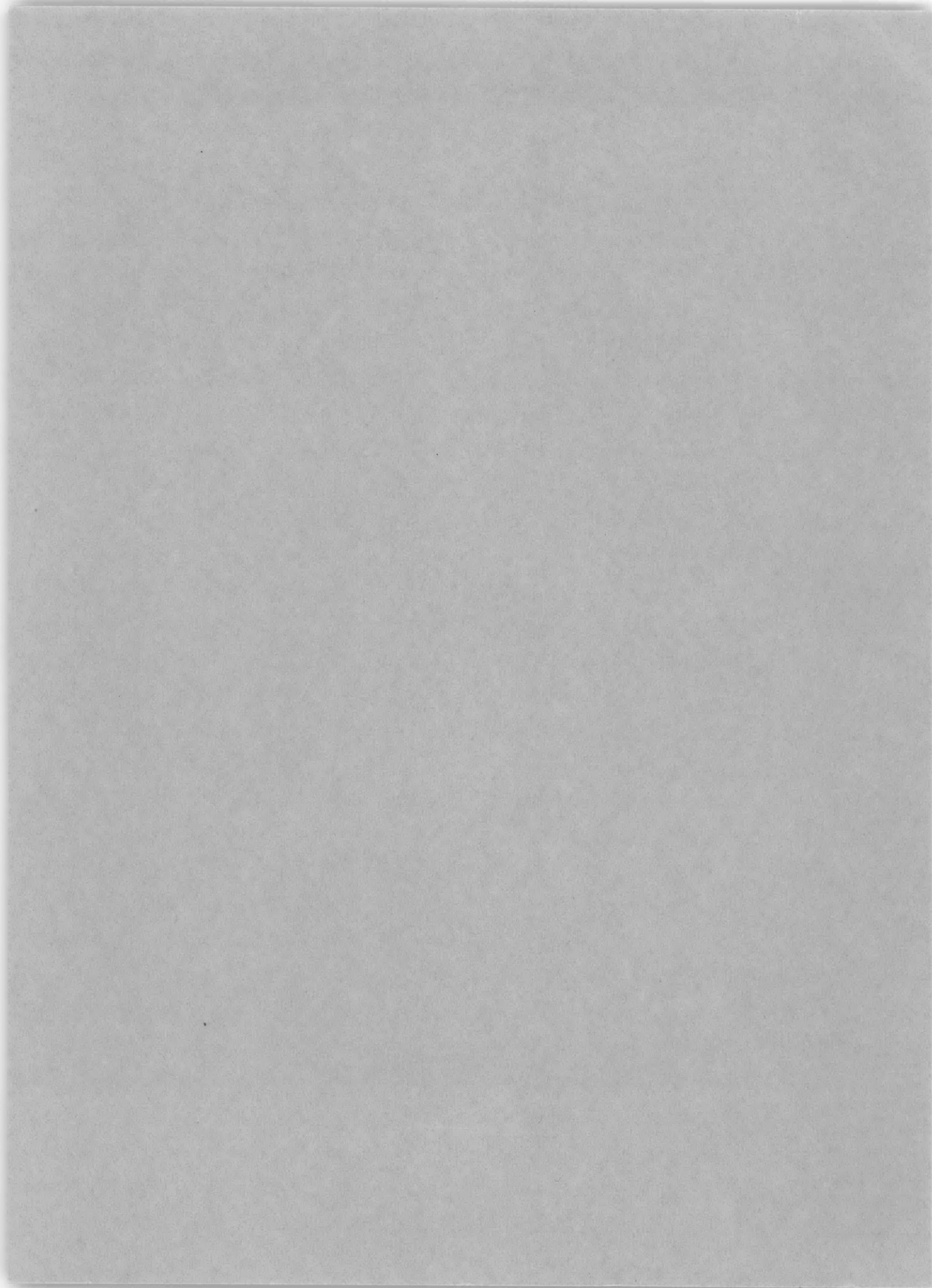
BY DR.-ING. C. KRUG



JULY 1942

RESTRICTED

TRANSLATION 101



LIGHTWEIGHT STEEL DESIGN OF MACHINE TOOLS

(STAHLLEICHTBAU BEI WERKZEUGMASCHINEN)

by

Dr.-Ing. C. Krug, VDI, Frankfurt a.M.

(VDI-Zeitschrift, 6 January 1940)

Translated by M.C. Roemer

The David W. Taylor Model Basin
Bureau of Ships
Navy Department, Washington, D.C.

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Translation 101

LIGHTWEIGHT STEEL DESIGN OF MACHINE TOOLS PRINCIPLES AND EXAMPLE

INTRODUCTION

While in many branches of engineering ultimate strength and work capacity (arbeitsvermögen) govern the dimensions of structural parts, particularly in the construction of vehicles, the design of machine tools is necessarily based upon the requirement of great rigidity. However, this by no means necessitates the use of large masses of material, but rather proper distribution of the material and correct design. The vibration of machines is not dependent upon the damping of the material to as great an extent as was long believed. Expertly constructed welded assemblies may be superior to cast-iron machines in respect to rigidity.

Following an outline of the principles of lightweight steel construction of machine tools, the advantages of this type of design are demonstrated by means of a particularly instructive detailed example. These advantages are maximum rigidity combined with light weight, and speed in manufacture.

RECENT DEVELOPMENTS

When lightweight steel construction of machine tools was first introduced in 1930 the question of freedom from shock was immediately raised, i.e., the question of how the new structures would withstand vibration. It was claimed that because of its excellent damping qualities cast iron was the proper material for machines under cyclic loads (1).*

During the years 1932 to 1938 a group of the leading machine-tool manufacturers, at the suggestion of the writer, sponsored tests in the Experimental Laboratory for Industrial and Machine-Tool Research of the Berlin Technische Hochschule, to clear up the basic problems of the dynamic behavior of steel-working machines in particular.

Kienzle's comprehensive report "On the Behavior of Lathe Beds of Lightweight Steel Construction," presented in March 1938 before a group of experts, showed that properly designed light steel structures were in no way inferior, as far as vibration is concerned, to similar cast-steel machines, and moreover, that a welded steel assembly has better damping qualities than a similar cast-steel unit. Furthermore, the tests showed that damping is not primarily a matter of material, but of suitable design.

The cited tests were continued by Kienzle and Kettner (2) and during the process the principles for dynamic testing of machine tools were established. The tests confirmed Krug's theoretical conclusions.

Kettner raises the basic question: Light or heavy machine tools? He distinguishes between "rigid light design" and "rigid massive design."

* Numbers in parentheses indicate references on page 13 of this translation.

For better understanding and more detailed explanation of his theories it will be desirable to discuss briefly the basic problem of the relation between rigidity, mass, and frequency of vibration, because inadequate views are still widely prevalent among practicing engineers as well as scientists.

BASIC PRINCIPLES

The basic equation for dynamic rigidity is

$$R' = \frac{P'}{a'} \sim f_0^2 m \left[\frac{\text{kg}}{\mu} \right] \quad [1]$$

where

- R' is the dynamic rigidity
- P' is the dynamic exciting force
- a' is the amplitude in resonance
- f_0 is the natural frequency
- m is the mass of the system

It is represented in Figure 1 by a family of hyperbolic curves. The line through the origin connecting the vertices divides the chart into two parts, I and II. Part I pertains to rigid light design, and Part II to rigid massive design (3).

RIGID LIGHT DESIGN AND RIGID MASSIVE DESIGN

For Part I slight changes in mass m cause large changes in the frequency f_0 . In accordance with the requirements of light design it is possible to achieve the requisite rigidity with relatively light masses. On the other hand Part II shows that

small changes in frequency can be accomplished only at the expense of large masses. The ratios become more unfavorable the greater the divergence from the requisite rigidity, i.e., the flimsier the machine.

It is also clear from the foregoing that the practice hitherto followed in building machine tools massive for rigidity is false and leads only to waste of material, and that in future another practice should be adopted, that of combining rigidity with light weight, long advocated by the writer (4).

As a rule the criteria in industrial design of structural parts are ultimate strength and load carrying ability, particularly in moving systems such

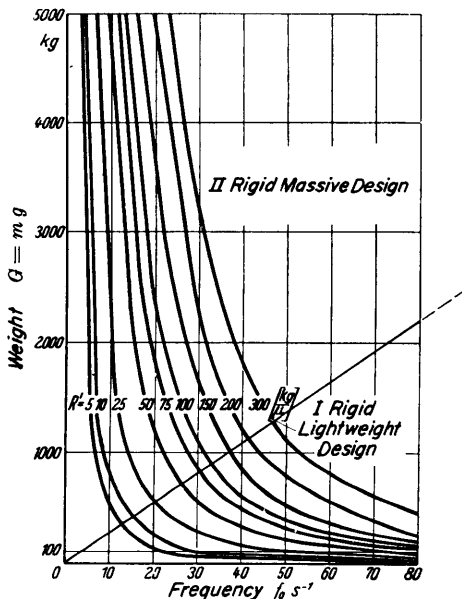


Figure 1 - Dynamic Rigidity $R' \sim f_0^2 m$

as railway rolling stock and motor vehicles, where the following relations apply:

$$\sigma_B = \frac{P}{a^2} \left[\frac{\text{kg}}{\text{cm}^2} \right] \text{ (Force per unit area)} \quad [2]$$

Here

- σ_B is the tensile strength in kg/cm²
- P is the load in kg
- a is the cross-sectional area under stress in cm²

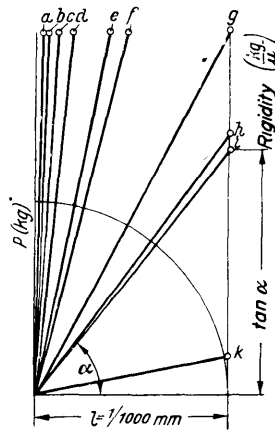
$$E' = P a \text{ [kgm] [Product of distance times force (5)]} \quad [3]$$

Here

- E' is the work or load carrying capacity of the system under load, measured in kgm
- a is the path on which the loading force P is stored up and released, in meters.

RIGIDITY OF MACHINE TOOLS

These two fundamental equations are of minor importance in the design of machine tools. The chief requisite of a machine tool is that it turn out parts accurately and in large numbers. This simultaneous requirement of output and precision can be satisfied only if no disturbing deformations of any kind occur at any point in the machine during operation, which might detract from the accuracy or neatness of the working surface. Heavy cuts to attain high output, can not be made on a flimsy, yielding lathe. If the piece of material yields on its path, more or less, or if the machine develops a chatter any-



- a Press and planer table
- b. Lathe bed 1
- c. Lathe head
- d. Box column of drill press
- e. Radial drill (box column)
- f Lathe bed 2
- g. Round column of radial drill
- h. Uprights of flat-bed grinder
- i. Bed of turret lathe
- k. Light drill press

Figure 2 - Rigidity R of Several Machine Tools
 $R = \tan \alpha$

where, it will be futile to expect a finished product accurate as to length and width, angle and area, let alone with a clean-cut surface. Therefore no disturbing elastic deformation, no springing of harmful proportions, can be permitted in machine tools. Such disturbing elastic deformations or whipping can be avoided only if the machine, in all its members, is as rigid as possible. A machine is rigid if it is able to absorb the greatest possible forces and transmit them, while undergoing the smallest

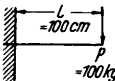
possible elastic deformations. Then the rigidity R , obtained from the deformation a in the direction of, and caused by, the static force P will be, in terms of μ








$$R = \frac{P}{a} \left[\frac{\text{kg}}{\mu} \right] \text{ (Force per unit length)} \quad [4]$$

Let $1 \text{ kg}/\mu$ be the unit of rigidity, i.e., a machine element which has a spring of 0.001 mm (0.000039 inch) under a load of 1 kg (2.2 pounds) has a rigidity factor of 1. This equation, as shown in Figure 2, represents a straight line through the origin, on which the rigidity is shown as a portion of the perpendicular lying at a distance $l = 0.001 \text{ mm}$ from the origin, which is cut off by any sloping line; the rigidity may also be represented by the angle of elevation α to the abscissa.

TABLE 1

Rigidity and Material Required

Load Application  Elastic Modulus = $E = 2.1 \cdot 10^6 \text{ kg/cm}^2$

Shape of Section	Moment of Inertia cm^4	Deflection cm	Rigidity kg/μ	Material Required	
				kg	Per Cent*
	24	0.66	0.15	13.6	100
	19.3	0.82	0.12	12.0	88
	38.4	0.41	0.24	5.9	43
	53	0.30	0.33	6.8	50
	68	0.23	0.43	4.7	35
	78	0.20	0.50	4.4	32
	68	0.23	0.43	4.2	31

* Values in this column represent percentages of the weight of a solid circular section, taken as the reference.

The angle of elevation α , when taken as the unit of rigidity, gives a clear idea of the rigidity of a structural element. If an element has a rigidity of 80 to 90 degrees it is evident that it has high rigidity, whereas a part with a rigidity factor of 10 degrees is considered as yielding.

Figure 2 shows the rigidities for a number of machine tools. The indicated values are derived from the literature and from tests made by the writer.

Kienzle and Kettner (6) extended the concept of rigidity and differentiated between compressive and tensile rigidity, flexural and torsional rigidity, according to the type of loading. Schenk (7) applied the theory of rigidity very effectively in the field of high-speed spindles.

The popular view that high rigidity involves large amounts of material is fallacious. High rigidity is by no means incompatible with small sections which represent economy of material (8). It is necessary only to abandon the use of solid sections, to arrange the material as far as possible from the neutral axis, and to make the height of cross section large in proportion to the free length. The latter measure should also be adopted to obtain high dynamic rigidity. Table 1, compiled for the case of bending with constant free length and load, with various sectional shapes and dimensions, shows the possibility of decreasing the weight of material and increasing the rigidity under the external conditions depicted in Figure 3.

MATERIAL ECONOMIES IN RIGID LIGHTWEIGHT DESIGN

The idea of lightweight design is based on the realization that it is possible to build structures which possess both lightness and rigidity. Depending on the type of material used, we speak of lightweight cast-iron structures, lightweight steel structures, and lightweight light-metal structures.

It is natural that special efforts should be made under present circumstances* to economize with the most important basic material, i.e., iron, and to take full advantage of the superiority in the strength of steel over that of iron. Since

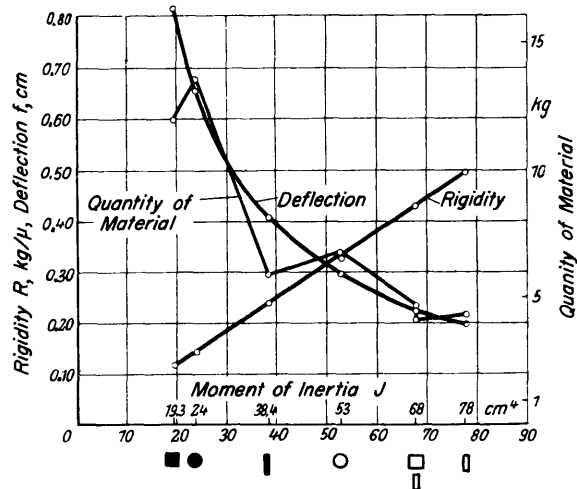


Figure 3 - Rigidity, Deflection, and Material Required for Sectional Forms shown in Table 1

* Translator's Note:- This paper was published in January 1940, when iron was a strategic material in Germany.

the elastic modulus of steel is approximately double that of cast iron* a saving of about 50 per cent in the quantity of raw iron will be possible by adopting lightweight design in steel, other conditions being equal.

Kienzle and Kettner, on the strength of years of experience in the study of machine tools at the Berlin Technische Hochschule, have stated that adequate static and dynamic rigidity can be attained in steel bases with a considerable saving of material. Moreover, they found that the objection originally made against the use of steel, that it is less suitable with respect to vibration, is untenable.

DAMPING OF MATERIAL

Engineers are still faced by many problems regarding vibration in machine tools, and particularly the effect of damping. Kettner casts some light on this subject in a series of important investigations and observations. The following are some of his basic conclusions:

1. The dynamic characteristics of a material, amplitude, frequency, and damping, do not alone govern the behavior of a machine as a whole in vibration.

2. The dynamic characteristics of a machine frame do not in themselves determine the vibrational behavior of the whole machine. Its behavior depends upon the interaction of these characteristics in the machine as set up and ready to operate.

These conclusions refute the frequently repeated statement that one material should be given preference over another solely because of better damping qualities. Machine damping is not a function of material, but primarily a function of design.

Kettner (9) states that when properly designed a structural-steel assembly has better damping than a cast-iron assembly, and that when improperly designed the steel structure may be inferior to cast iron.

The following relations apply to machine-tool design:

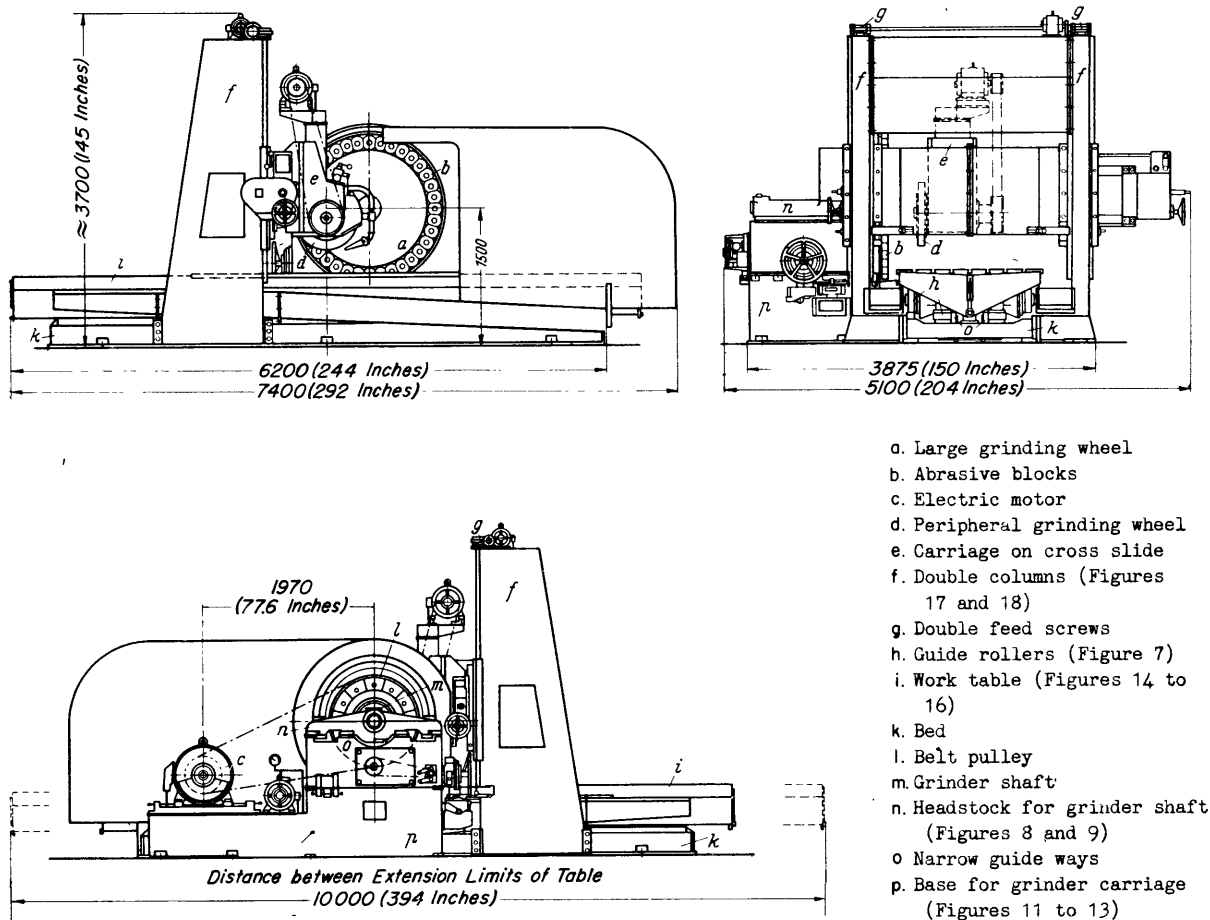
1. A statically weak body usually is dynamically weak also. However it may act as a dynamically rigid body under forced high-frequency vibrations.

2. A statically rigid body usually is dynamically rigid also, but not necessarily so; in the resonance zone it becomes weak unless damped.

3. Dynamic rigidities vary as the squares of the natural frequencies; for example, if the natural frequency is doubled, the dynamic rigidity will be quadrupled.

4. Deflections vary inversely as the squares of frequencies. To illustrate, if the frequency is increased from 5 CPS to 50 CPS, the deflection

* The modulus of elasticity gives the rigidity per unit volume of a material.



Figures 4 to 6 - Large Grinding Machine of Lightweight Steel Construction

The segmental grinding wheel *a* is capable of grinding the side of the work over its entire depth; at the same time the peripheral abrasive wheel *d* can grind plane surfaces as well as depressions such as keyways and slots. The feed for both wheels is hydraulic or manual. The transverse carriage *e* is adjusted vertically through the electrically-driven twin feed screws *g* and a worm gear. The feed of the work table *i* is hydraulic and infinitely variable.

Diameter of segmental wheel	1600 mm 63 inches	Floor space required	10000 mm x 5100 mm
Diameter of peripheral grinder	700 mm 27.5 inches		394 inches x 204 inches
Longitudinal capacity	2500 mm 98.4 inches	Automatic hydraulic or manual control of cut, in	steps of 0.01 to 0.1 mm (0.00039 inch to 0.0039
Lateral capacity	1400 mm 55.2 inches		inch) for a total depth of 75 mm (2.95 inches)
Vertical capacity	1400 mm 55.2 inches	Total weight	about 22000 kg (48500 pounds)
Working area of table	1600 mm x 3000 mm	Total power required	about 76KW
	63 inches x 118 inches		

will decrease to 0.01 of its original value; for instance, from 1 mm to 0.01 mm (0.039 inch to 0.00039 inch).

5. Dynamic rigidities vary inversely as the square roots of the masses; if the vibrating mass is quadrupled, the natural frequency decreases by half.

6. Amplitudes vary inversely as the vibrating masses; if the vibrating mass is reduced by one half, the deflection will be doubled.

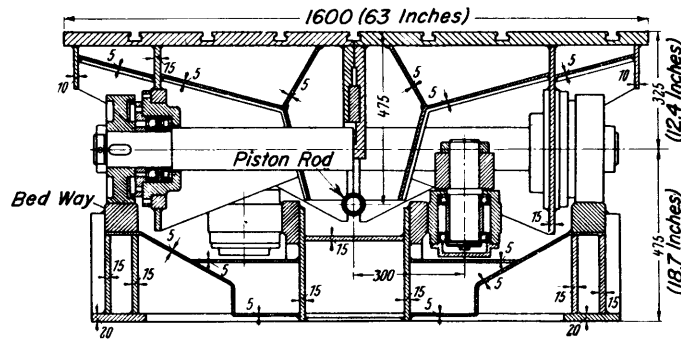


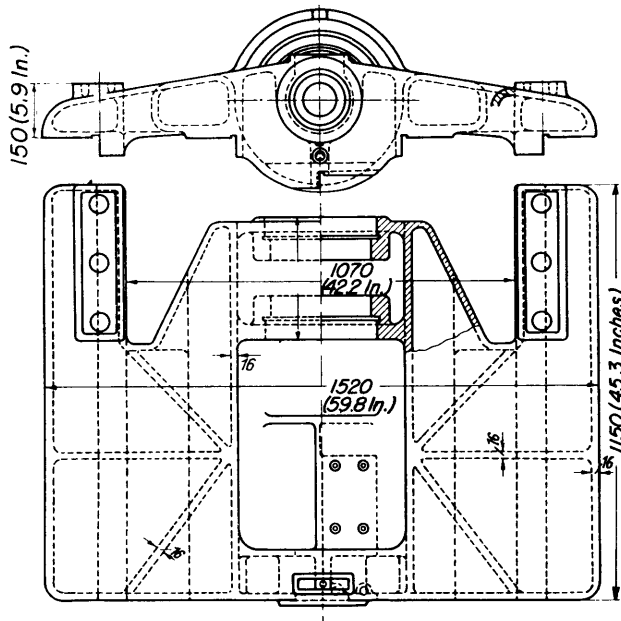
Figure 7 - Section through the Roller System of Table and Bed

The table moves on the bed ways on four vertical rollers, mounted in pairs on two ball-bearing horizontal shafts. Lateral guiding is done by four horizontal rollers which are likewise mounted on ball bearings on vertical shafts. This arrangement reduces friction between table and bed. The numbers are dimensions in millimeters.

ELIMINATION OF VIBRATION IN MACHINE TOOLS

It is important in machine tools of all designs to eliminate harmful vibrations. These usually result from unbalance of rapidly revolving parts, especially in drive or auxiliary motors, and irregularities in the gears and ball bearings.

To eliminate vibration in machines it is best to measure the amplitude with a vibrometer* and determine the direction and frequency of vibration. This done, it is easy to find the source of the disturbance, to measure the amount of unbalance with the vibrometer, and to balance it. It is also possible to damp out any other sources of vibration and to investigate whether the material is distributed properly, and more particularly, whether more reinforcing ribs are required. The vibrometer is easy to operate, fast, and gives highly accurate data.



Figures 8 and 9 - Headstock for Grinder Shaft in Lightweight Cast-Iron Design

Compact design with stiffening ribs properly arranged to take up the bearing forces. The wall thickness is 16 mm (0.63 inch)

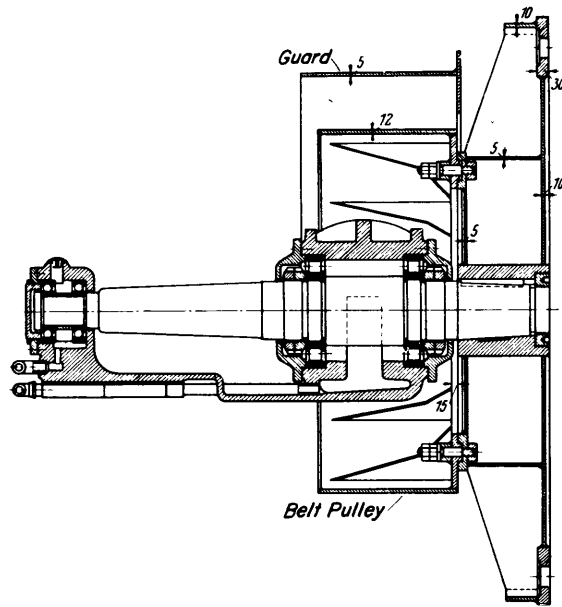
INTERNAL STRESS AND ANNEALING

There are many inconsistencies in the methods of treating the problems of internal stress and

* The vibrometer of Dr. Haymann, Darmstadt, is used in the Diskus Werke Maschinenbau A. G., Frankfurt a. M.

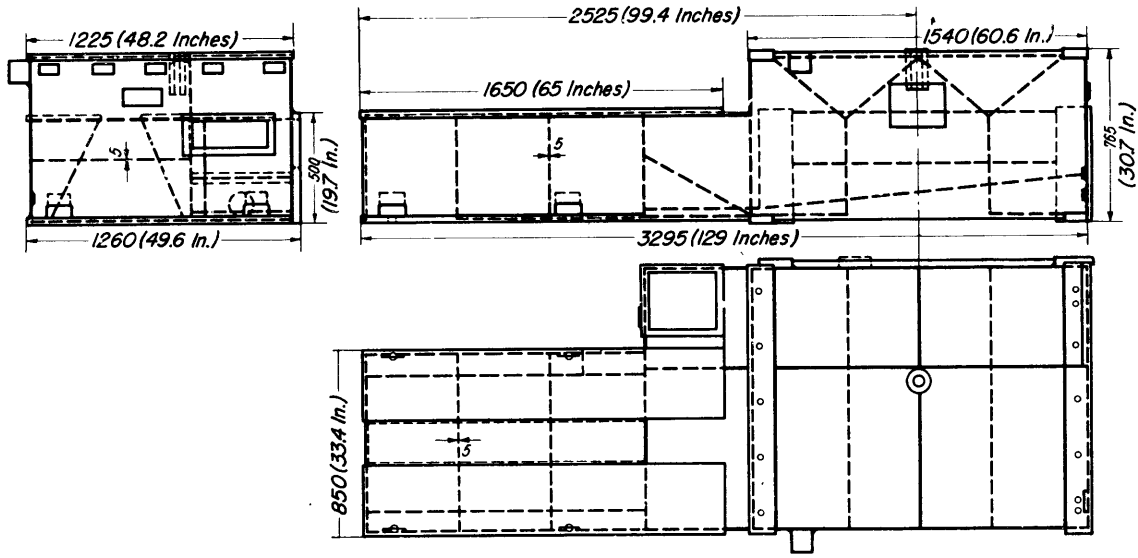
Figure 10 - Mounting of Shaft of Side- or Face-Grinding Wheel

The heavy shaft has a diameter of 185 mm (7.28 inches) with two roller bearings and two Radiax bearings. By direct connection of the belt pulley with the face plate of the large grinding wheel the torque is transmitted directly to the wheel without first being led through the shaft. The forces of the belt drive are taken up directly by the roller bearings, which are placed close to the pulley. The belt pulleys and supporting discs are built up of welded steel plates.



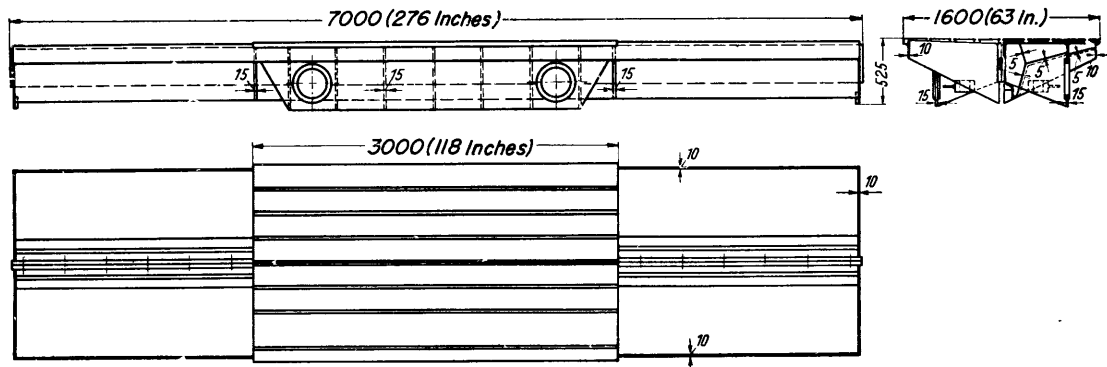
of annealing. In cast-iron and steel structures internal stresses are caused by heat. In both cases the shop will have to adopt measures to render the effect of such stresses negligible in practice. Harmful deformations due to internal stresses caused by electric welding can be prevented by expert workmanship, or if they have already occurred, can be corrected easily. Superficially the stresses are not evident since they are in equilibrium. But if the equilibrium is disturbed, perhaps by the process of fabrication, the same remedial measures should be applied as are used with cast iron. At all events, structural steel technique has so far advanced that in general it is no longer necessary in machine building to anneal parts after completion in order to equalize welding stresses in the manufacture of the main parts, such as beds, frames, tables, uprights, bases, arms, yokes, and the like. Moreover, there is no cause to fear subsequent bending or warping with the passage of months or years. Considerable data are now available on that subject with regard to precision tools. It is requisite that the thickness of the plating used should not exceed a given value. However, greater thicknesses generally are not needed, since methods have now been developed of building machine frames of high rigidity and strength by welding together relatively thin plates, without danger of harmful vibrations. Even though errors are still made occasionally, the rapid progress in structural engineering will soon eliminate them.

With regard to the accuracy with which electrically welded parts of machine tools can be made to conform to the working drawings, a deviation of 1 mm per meter (0.001 inch per inch) may be expected, although this value has already been cut down considerably. On the average, the machining allowances applied to welded or riveted steel structures will be about half the amounts necessary with cast iron.



Figures 11 to 13 - Frame of Side Grinder Carriage, including Base for Drive Motor
 The frame is connected directly to the grinder carriage and is designed as an oil tank. It carries the 75 HP drive motor as well as the oil pump and the pump motor.

It is difficult to compare the relative economic efficiencies of cast-iron and steel structures. This question is widely discussed in technical circles. While steel is now generally preferred for building machines singly or on special designs, cast iron is considered as better adapted to mass production methods. The reasoning and calculations leading to this belief are correct so long as the material involved is available cheaply and in unlimited quantities. But when this is not true, conditions may rapidly change in favor of steel construction which brings about great savings of material. If in addition a considerable saving of construction time can be achieved by the use of steel, it is possible that cast iron will have to yield its place in the industries involved.



Figures 14 to 16 - Work Table

The work table is built in two parts bolted together along the longitudinal centerline. The two ends of the table act as covers for the roller beds and as containers for the cooling agent.

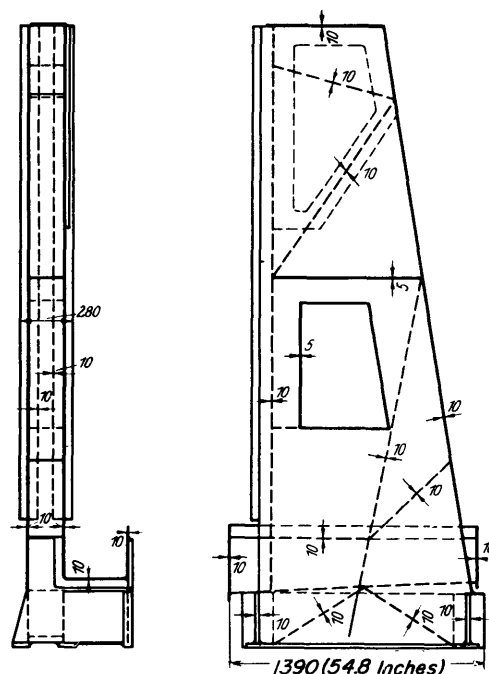
Obviously there may be cases in which the opposite is true; for example when rolled steel or electric power for welding are not readily available, castings would have to be used.

The use of lightweight steel construction for machine tools has been discussed repeatedly in the recent literature (10). The success or failure of a steel structure depends primarily on the ability of the designer to disregard casting technique entirely. In particular he must not let himself be bound to retain previous structural dimensions or to adhere to existing fits to permit use of available structural members, such as tail stocks or head stocks of lathes, and tables or drives of milling machines.

EXAMPLE OF APPLICATION

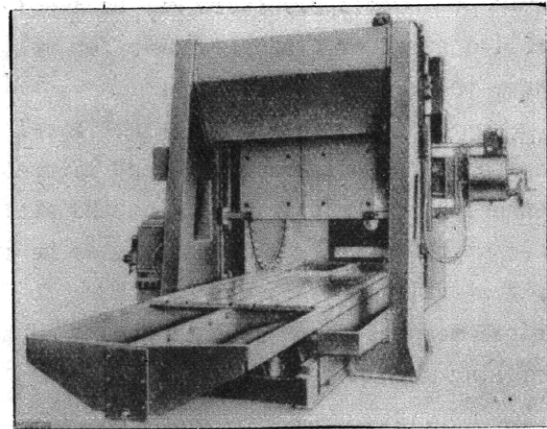
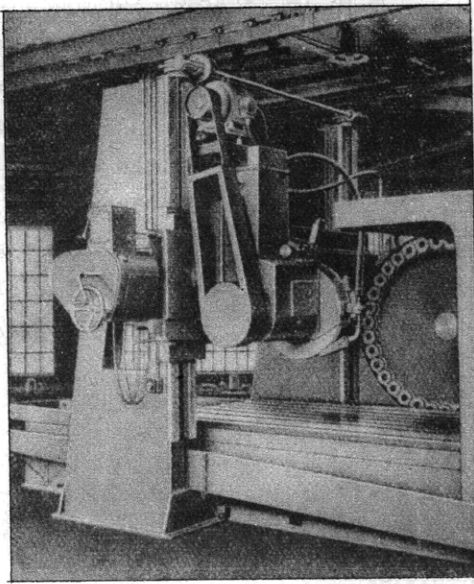
To refute the objection that lightweight steel design is not suitable for large machine tools, an example is cited of its application in that field.

The machine depicted in Figures 4 to 6 was built to grind to an exact plane pieces of work weighing 5,000 kg to 30,000 kg (11,000 to 66,000 pounds), up to 2,500 mm (98 inches) in length, up to 1,400 mm (55 inches) in width, and up to 1,400 mm (55 inches) in height, with an accuracy of ± 0.03 mm (± 0.001 inch). It has a lateral feed against the large wheel *a* of 1600 mm (63 inches) diameter, which is equipped with a circular row of individual face grinding blocks *b*, and is driven by a 75 HP electric motor *c*. The work can be ground flat simultaneously in a horizontal plane by means of a vertical wheel *d* with a grinding face around its periphery. This wheel has a diameter of 700 mm (27 1/2 inches) and a width of 80 mm (3 inches). This peripheral wheel is mounted on a sliding carriage *e*, which can be moved vertically or horizontally, as in a planer. The table *i*, with a work area of 1600 mm by 3000 mm (63 inches by 118 inches), is driven hydraulically. The peripheral grinder is also moved across the direction of feed hydraulically, and its depth of cut is regulated by two electrically-driven feed screws *g*. The total power input for the machine is about 100 HP. The total weight of the machine is approximately 22,000 kg (48,500 pounds); in cast iron, it would have been about 50,000 kg (110,200 pounds).



Figures 17 and 18 - Design of the Two Columns or Standards

The double standard is built in cells of welded steel plating 10 mm (0.4 inch) thick, and carries the cross carriage or bridge.



Figures 19 and 20
Views of the Grinding Machine

PECULIARITIES OF THE DESIGN

1. Inclusion of a grinding wheel *a* for grinding a vertical side of the work and a wheel *d* with peripheral grinding for a horizontal surface.
2. New steel-reinforced grinding wheel with peripheral cutting surface which assures full use of abrasive material.*
3. Work table mounted on roller bearings *h*, because of the great weight of the work.
4. Reduced height of the bed *k*, and correspondingly increased height of the table for rigidity.
5. Direct drive of the grinding wheel through attached large belt pulley *l* which is clamped on a flange of the wheel without loading the shaft *m*.
6. Use of lightweight iron castings for the extremely low, squat grinder-shaft headstock *n*. In this case castings are preferred, owing to the complicated, cramped design.
7. Narrow guide ways *o* for the table and the grinding wheel carriage.
8. Electrically welded steel plate construction.** Plate thicknesses in the base were 3 and 5 mm (0.12 and 0.2 inch), in the uprights 5 and 10 mm (0.2 and 0.4 inch), and in the bed 5 to 15 mm (0.2 to 0.6 inch).

* German Patent 661807.

** German Patent 755939.

In the shop tests the table surface was found to be accurately machined to ± 0.01 mm (0.0004 inch) and tests of side deviation showed an accuracy of under 0.01 mm (0.0004 inch). Figures 7 to 20 show the most noteworthy features of the grinding machine, and the legends below the figures give the details.

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- (8) See A. W. F. *Mitteilungen*, vol. 21, 1939, pp. 97 to 99.
- (9) See Reference (2), p. 16, column 2, section 4.
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