

FIG. 7.21

### Spring Design

Whilst there are numerous formulae, empirical and otherwise, governing the design of helical compression springs such as are employed in safety valves, the rules prescribed by the Board of Trade should be rigorously observed. These rules are amplified in B.S. 759: 1950 *Valves, Gauges and Other Safety Fittings for Land Boiler Installations* to which reference has previously been made. According to these rules, the initial compression to give the desired load is one-quarter the diameter of the valve seating but the Admiralty specify double this amount.

Much confusion regarding the lift of a safety valve when blowing off at full can be attributed to a misinterpretation of the term *compression* in the Board of Trade formulae. This specifically refers to the *initial compression* to be imparted to the spring in order to produce the necessary axial force to counter the pressure upthrust and other resistances and does not apply to any *subsequent compression* resulting from valve lift, which is very small in comparison.

Clause 26 (b) of B.S. 759: 1950 states—

*The compression or extension of safety valve springs required to load the valves to the set pressure shall not be less than one-quarter of the diameter of the valve. . . .*

Whilst this clause states that the initial compression (or extension) shall *not be less than* one-quarter of the diameter of the valve, this value is rarely exceeded by designers in their calculations as the stress induced in the material of the spring is directly proportional to the compression (or extension).

Clause 26 (b) continues—

*The proportion of unloaded length to external diameter of the spring shall not exceed 4 to 1.*

This calls for no interpretation although many designers fall into the error of making the free length of compression springs very much in excess of  $4D_o$  (see Fig. 7.23) resulting in a spring which is lacking in lateral stiffness, and tending to become bowed. This tendency may cause tilting of the valve member and undue flexing of the spindle whose thrust should always be co-axial with the valve member if freedom of movement of the parts is to be assured.

This error may generally be attributed to the number of inter-dependent variables in the spring formulae which often involve the designer in much tedious repetitive calculation.

Clause 26 (b) also states—

*The maximum shear stress as determined by the following formulae shall not exceed 80,000 lb/in.<sup>2</sup>*

(i) Round Section

$$\text{Stress} = K \frac{16SR}{\pi d^3} C \quad \text{Eq. 7.11}$$

(ii) Square Section

$$\text{Stress} = K \frac{4.8SR}{d_s} C \quad \text{Eq. 7.12}$$

(iii) Rectangular Section

$$\text{Stress} = K \frac{(3B + 1.8H)SR}{B^2H^2} C \quad \text{Eq. 7.13}$$

where

$$K = \frac{\frac{4D}{d} - 1}{\frac{4D}{d} - 4} + \frac{0.615}{\frac{D}{d}} \quad \text{Eq. 7.14}$$

(In the case of rectangular sections substitute  $B$  for  $d$ .) $S$  = load in pounds at set pressure $R = \frac{D}{2}$ , mean radius of coil (inches) $d$  = diameter of round or side of square steel (inches) $B$  = breadth of wire (radial to spring axis) (inches) $H$  = depth of wire (parallel to spring axis) (inches) $D$  = mean diameter of coil (inches) $C$  = constant

$$= \frac{L_1 \times L_2}{L_1}$$

$L_1$  = initial compression or extension of the spring (in.) to the required loading ( $P \times A$ ) where  $P$  = design pressure (lb/in.<sup>2</sup>) (set pressure);  
 $A$  = loading area of valve

$L_2$  = the further compression or extension of the spring to give the lift as defined in Clauses 13 and 25 (in inches)

Examples

$C = 2$  where compression or extension of spring to give the required loading is  $\frac{1}{4}$  diameter of valve

$C = 1.5$  where compression or extension of spring to give the required loading is  $\frac{1}{2}$  diameter of valve

$C = 1.25$  where compression or extension of spring to give the required loading is full diameter of valve

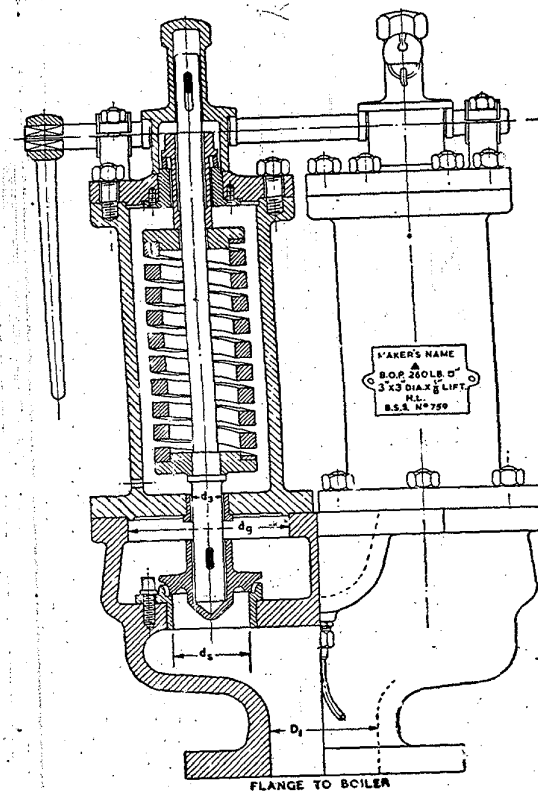


FIG. 7.22

$D = \sqrt{2} d = 1.415d$  for double valve

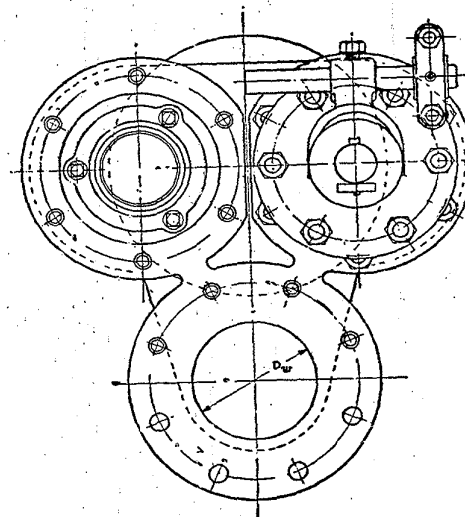
$D = \sqrt{3} d = 1.732d$  for triple valve

$D = \sqrt{4} d = 2.000d$  for quadruple valves

$D_w = \sqrt{\frac{2}{P\pi}}$  for ordinary valves

$D_w = \sqrt{\frac{2E}{3\pi P}}$  for high-lift valves

$D_w = \sqrt{\frac{4E}{9.6\pi P}}$  for full-lift valves



In order to simplify the evaluation of the oft recurring ratio of mean diameter to size of coil ( $K = \frac{D}{d}$ ), values of  $K$  may be determined from the graph shown in Fig. 7.24.

For small valves and moderately low pressures, springs of round-section wire are usually employed, square section being reserved for the larger sizes of valves or those intended for high pressure, whilst springs of rectangular section are specifically intended for valves having high lift characteristics in order to provide for more than the usual amount of subsequent deflexion.

Clause 26 (e) states—

... The number of effective or free coils in a compression or extension spring shall be determined from the following formulae—

$$(i) \text{ For round or square wire } N = \frac{KCd^4}{SD^3} \quad \text{Eq. 7.15}$$

$$(ii) \text{ For rectangular wire } N = \frac{66B^3H^3K}{(B^2 + H^2)SD^3} \quad \text{Eq. 7.16}$$

where  $N$  = number of effective coils

$K$  = compression or extension in inches at set pressure

$C$  = 22 for round, 30 for square steel

$d$  = diameter or side of square steel in 16ths of an inch

$S$  = load on spring in pounds at set pressure

$D$  = mean diameter of coil in inches

$B$  = breadth of wire in 16ths of an inch

$H$  = depth of wire in 16ths of an inch

Having satisfied the requirements stipulated in Clause 26 (b), namely, that the shear stress in the material of the spring does not exceed 80,000 lb/in.<sup>2</sup>, the number of effective coils may now be determined from the above formulae.

By effective coils is meant the actual number of coils which take part in the resistance of the applied loads and not the dead coils, or portions of coils, at each end of the spring. The latter, being squared off and ground flat to present suitably flat bearing surfaces (see Clause 26 (g)) cannot be relied upon to take any share of the load. Consequently, these dead coils usually totalling from 1½ to 2 in any compression spring, must be discounted from the number of complete sections. This is best illustrated in Fig. 7.23 (see part-sectional elevation) in which the number of effective coils may be taken to be nine.

Clause 26 (f) states—

The space between the coils, when the valve is lifted one-fourth of its diameter, shall not be less than  $\frac{1}{16}$  in.

Having determined  $N$ , the number of effective coils, it is an easy matter to assign a suitable overall length to the spring based on the value of  $N$  thus determined, and having regard to the minimum allowable space between the coils called for in the clause, allowing

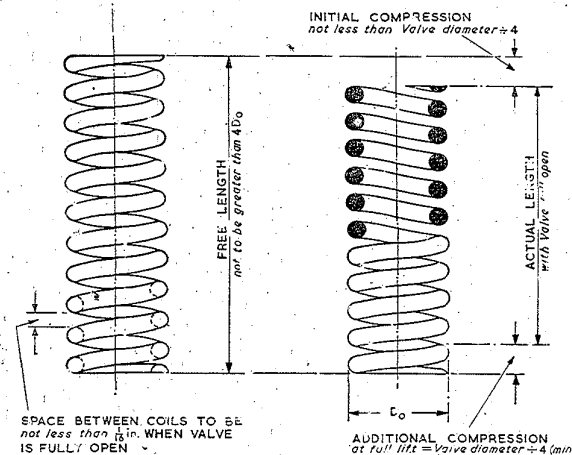


FIG. 7.23. SAFETY VALVE SPRINGS

for the number of dead coils and the initial and working deflexions. There then remains the final task of checking the dimensions to ensure that the ratio prescribed in Clause 25 (b) is obtained. If not, then some amount of adjustment will be necessitated until a satisfactory relationship is obtained.

Clause 28 states—

Each valve shall be clearly and legibly marked with the following—

- Manufacturer's name and identification mark.
- Set pressure in lb/in.<sup>2</sup>
- Diameter and design lift in inches. If High Lift the letters HL and if Full Lift the letters FL to be added.
- The number of this British Standard (B.S. 759.)

The design shown in Fig. 7.22 incorporates an inscription plate of a type which complies with the above Clause. Such inscriptions may be cast in raised letters and numerals on the body or otherwise conspicuously displayed, or may be arranged on a separate plate securely attached to the valve.

Fig. 7.22 (and also Fig. 7.18) depicts a typical double-spring safety valve which is generally styled "Marine Pattern" since it is favoured for use on marine boilers. Whilst the design portrayed does not represent any particular maker's standard nevertheless it is designed in strict conformity with existing rules governing the design of such a valve. Triple, or even quadruple, valves of similar construction to the one shown are sometimes called for, although the double valve is the one more commonly employed, the high lift characteristics of certain makes rendering anything more than two valves superfluous.

The main inlet thoroughfare will naturally require to have an area at least equal to the combined areas of the separate valve seatings whence it can be shown that the diameter  $D_1$  of the inlet thoroughfare will be given as\*

$$D_1 = \sqrt{2} d_s = 1.414 d_s \text{ for double valves} \quad \text{Eq. 7.17}$$

$$= \sqrt{3} d_s = 1.732 d_s \text{ for triple valves} \quad \text{Eq. 7.18}$$

$$= \sqrt{4} d_s = 2.000 d_s \text{ for quadruple valves} \quad \text{Eq. 7.19}$$

### Chests

The complex shape of the chest defies any attempt to determine accurately the thickness, but the treatment given in Chapter 15 might be applicable if due discretion is exercised and experience and precedent accepted as the main guiding factors. A liberal thickness is advocated in the region of the neck to cater for the effects of vibration in addition to that of meeting pressure requirements. For the same reason the inlet flange should be made larger than that appropriate to the pipe size denoted by the bore  $D_1$ .

The chest is usually made in cast iron for all steam pressure up to 240 lb/in.<sup>2</sup>, saturated, but the general requirements prescribed in B.S. 759: 1950, Clause 2 (a) *et seq.* should be observed. (Reference will be made to this aspect in Chapter 15.)

It is suggested that cast steel be adopted wherever there is any doubt regarding the choice of material for the valve chest.

The spring casing may be of cast iron and it will be appreciated that this will be subjected to a direct tensile force resulting from the spring thrust and transmitted by the compression screw and top plate to this member. Basing design solely on the tensile force imparted invariably results in a relatively thin wall thickness, but  $\frac{3}{8}$  in. should be regarded as a minimum value if only from considerations of obtaining sound castings.

\* It is usual to round off  $D_1$  to the nearest standard pipe size.

If  $C_m$  is the mean diameter of the spring casing then the thickness  $t_c$  will be given very approximately by the relation

$$t_c = \frac{F_a}{\pi C_m f_t} \quad \text{Eq. 7.20}$$

where  $f_t$  is the allowable tensile stress, say 3,000 lb/in.<sup>2</sup> for cast iron,

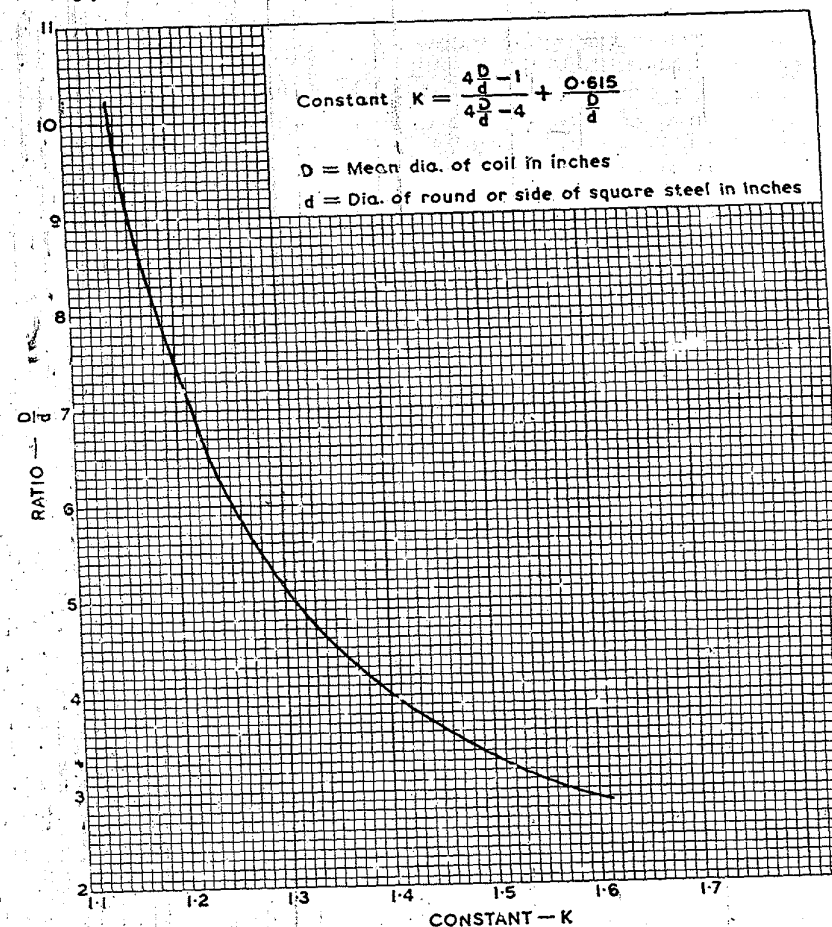


FIG. 7.24

but the ratio of area presented to that of the axial load imparted is generally so high as to give a thickness, by the exploitation of this formula, too small to satisfy practical requirements.