A STUDY OF EXPANSION JOINTS FOR STEAM RISERS

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INTRODUCTION

In view of the fact that all metal conduits, experiencing the changes of temperature that are inherent in steam transportation, require the use of some form of expansion joint, it should be of interest to those in the engineering profession, or those about to enter that profession, to become acquainted with the general characteristics of expansion joints for steam pipes. With this idea in mind, this paper was written.

The most important factors to be considered in this case are, first of all, the general treatment and purpose of the expansion joint. The principles of the different types of expansion joints, and their limits in use, are discussed briefly. The greater portion of the space is given to tests and operations of the packless type of joint manufactured by The Fulton Sylphon CO., Knoxville, Tennessee. Included in this paper are photographs of testing machine and different kinds of expansion joints, data obtained from test, and conclusions derived from data.

It would have been hardly possible for the compilation of this material had it not been for the engineering files of the Fulton Sylphon Company. I am also greatly indebted to the engineering librarian of the University of Tennessee, Miss Laura Kersey, for her aid in securing material for me on this subject.

GENERAL SURVEY OF THE FIELD

A few years ago,when building materials and labor cost about half as much as now,tall buildings were built with a liberal use of space that is seldom found today. At that time the usual way of carrying for the expansion of heating risers was to provide "expansion loops" of pipe at any desired level,concealing them in hung ceiling spaces.

Present day building costs have virtually revolutionized space conditions,making impossible the former methods of absorbing expansion and contraction of risers and returns. Fifteen stories are crowded into a height of building that formerly accommodated eleven or twelve floors.The depth of fill above floor arches has been reduced. Hung ceilings are a rarity and in many buildings the bottom of the floor arch is so near the window heads that any sort of ceiling pocket or false beam to conceal expansion loops is out of the question.

Obviously the answer is some sort of joint,inserted right in the vertical riser. Such a joint has the advantage of using a complete manufactured unit instead of assembling a substitute by labor on the job. At present wage rates the labor item for assemblying a large expansion loop is considerable enough in most cases to pay for a good expansion joint. The manufactured joint can be installed in the lines as simply as any other single pipe fitting.

To meet these modified conditions of building construction, packed, or "sliding sleeve" expansion joints were for a long time the only resort for designers. They meet the problem of space conditions in the case of exposed risers, but where the joints must be concealed in the walls it requires much expense and labor to get to them for repacking. The packed joint may be loose enough to leak or tight enough to jam. It must be repacked from time to time, and this is an almost impossible job in the case of a riser concealed in furring.

The development of this situation has pointed clearly to the need of a packless joint which would at all times be steam tight without any necessity for attention and at the same time have perfect freedom of motion without risk of jamming. All of this is supplied in the packless bellows type joint.

The selection of an expansion joint for a definite job today is not made at random, but by considering the type of joint that will give the best service on that particular use. There are three general means used to compensate for expansion and contraction in steam pipes: the expansion loop, the slip-tube packed joint, and the packless corrugated joint. The expansion loop has almost entirely gone out of use but it is yet sometimes used where there is ample space and its bulk does not distract from the looks of anything, or where no one is likely to see it. The slip-tube packed joint is widely used because they are suitable for high pressures and all sizes of pipes. They require no more space than an ordinary fitting such as unions, couplings, and ells. They are so designed that they can be repacked from time to time, and they may be used for a life time of a building without being removed. They are designed in many different mechanical set-ups

since they are manufactured by many different companies. Some joints are so designed as to be externally guided and others are so designed as to be internally guided as the available space demands. They are all packed essentially a like and their degree of goodness or poorness is determined by the kind of packing used, means of guiding, and the anchoring of the pipe line. The most essential requisite of any set-up is that the joint be in true alignment with the pipe line so that the slip will be such as to have stresses only between the tube and the packing.

The packless corrugated expansion joint has a definite field in which it can safely be applied. This is for low pressures and for vertical lines. They have been employed for pressures up to 200 pounds per square inch by using reinforcing rings to support the corrugations. Similar to the other type there are many designs of packless corrugated joints made, but the principle in each case is the same. The expansion is taken care of by deep corrugations in copper tubes. They are designed for small expansions, usually not over 1-1/2 inches. Since this type of joint contains copper bellows they cannot be used in lines carrying superheated steam, due to the corrosive action of the steam upon the copper. The main advantage offered for thie type of joint is that no packing is required so that it can be installed in a wall or hemmed in any place without providing a means of acces to it. Inspite of this good quality it has been discovered from experience that it is never advisable to install any joint without providing an opening to it so that it can be repaired or replaced without having to tear out a wall or floor which would be more expensive than making means

of access for the joints.

If satisfactory service is expected of any type of expansion joint it must be properly installed in a line that is securely anchored so that the joint will only have to take care of the expansion that it is supposed to absorb and that no side thrusts are thrown into the joint which might cause it to jam, and as a result, fail when the temperature rises.

The field of expansion joints is very highly developed due to the keen composition that has arisen between the many manufacturers of them during the last few decades of skyscraper building programs. The greatest need in the development of the packless corrugated type of joints is a metal, from which to make the bellows, that has a higher resistance to steam and its impurities for a longer period of time than the present metal used has. No revolutionary changes in the present designs are anticipated.

STATEMENT OF PURPOSE

The purpose in undertaking this work was to make a general study of the field of expansion joints and, at the same time, to make a more complete study of the Sylphon packless expansion joint. The central idea in the general study of expansion joints is to determine the adaptability of the many kinds of joints to the particular field for which they are designed. The purpose is to enable one to select the proper joint for the most effective service in the fields of steam that an engineer is likely to encounter.

A more complete study is made of the packless corrugated expansion joint in order to determine the most probable steps to take in improving the efficiency of the present design. The tests will be made on the Sylphon packless joint manufactured by the Fulton Sylphon Company of Knoxville, Tennessee. Also, a test will be made on a packless corrugated joint manufactured by the Ross Heater and Manufacturing Company of Brooklyn, New York.

GENERAL THEORY

Any pipe line subject to variations in temperature undergoes a proportional change in length. The increase in length may be expressed in the following equation:

$$I = aL_0 t$$

in which I is the increase in length, a is the coefficient of linear expansion, L_0 is the original length, and t is the rise in temperature. The coefficient of linear expansion of any substance is the increase in length of unit length for one degree of rise in temperature. With zero as a base temperature and the equation above, values were calculated and the curves plotted as given in Figure 1. These curves indicate the amount of expansion per one hundred feet of length of wrought iron or steel pipe carrying either saturated or superheated steam.

It has been discovered from experiment that different metals expand at different rates, that is, their coefficients of linear expansion is not the same. It has also been found that the coefficients of linear expansion for wrought iron and steel are practically the same, which is 0.0000116 inch per degree centigrade.

Various methods of compensating for this contraction and expansion have been provised. Chiefly among these are the expansion loop, the slip-tube packed expansion joint, and the packless corrugated expansion joint. Compensation with the expansion loop involves the use of a specially constructed bend inserted in the pipe line at the point where the expansion is to be absorbed. Preferably, the bend is made of a single length

of pipe and the radius should be at least six times the diameter of the pipe. This means that the " U " bends of a single length can be used with pipes 12 inches in diameter or less since pipes usually do not come in lengths over twenty feet. With pipes larger than this two or more pipes are required and they are joined together at the middle of the bend or at the two corners of the " U ". These joints are inevitably subject to excessive strain, and it is often difficult to keep them tight. For constructional purposes the tops of the loops should have straight runs of at least two times the diameter of the pipe. Vertical bends tend to form pockets in which condensate collects if due provision is not made for draining. When expansion and contraction is compensated for by loops, anchorage should be extremely heavy and rigid since there are enormous stresses in the pipes. A picture of an expansion loop is shown in Figure 2.

The slip-tube expansion joint consists of a sleeve closely fitted to a body and surrounded by a deep stuffing box. The underlying principle is that no transverse stresses are allowed to be transmitted to the stuffing box. The guides are entirely independent of the stuffing box and the only stress to which the packing is subject is a direct compression by the packing box follower and the pressure in the line due to the steam. The efficiency with which a slip-tube joint compensates for expansion and contraction in steam pipes depends almost entirely upon the guides, the anchors, and the packing. All

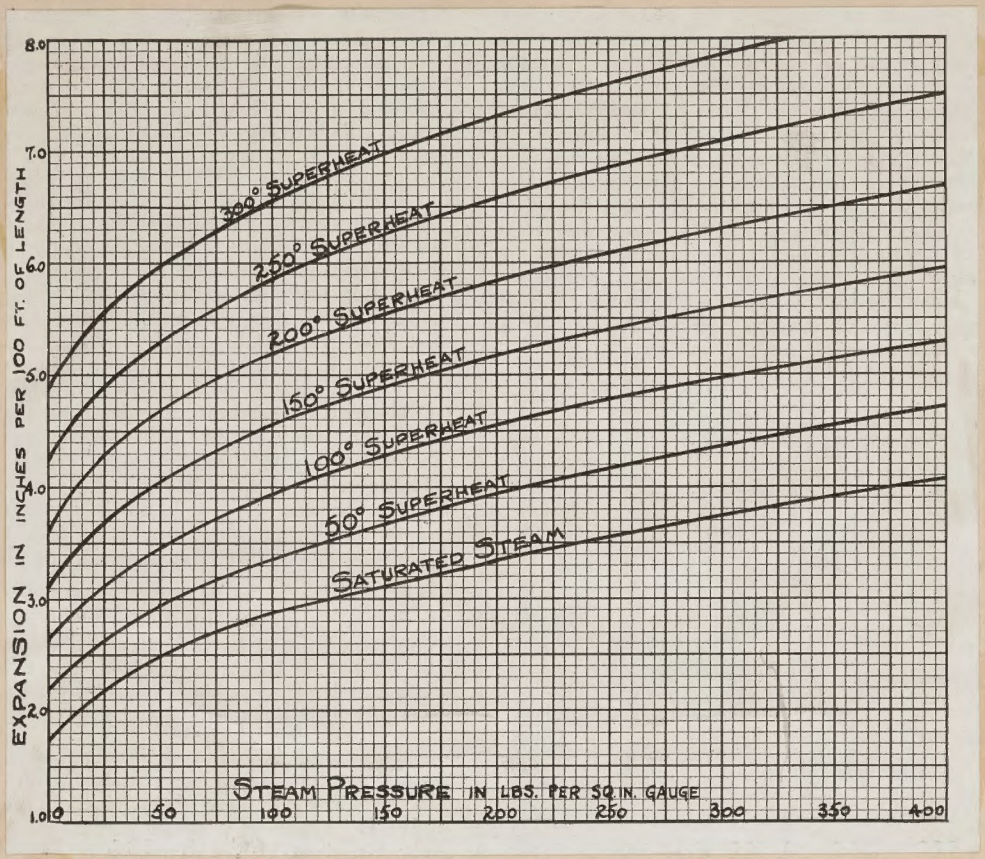

Figure 1. Compensating For Expansion in a Pipe Line



Figure 2. Expansion Loops

three of these are very well taken care of in the present state of development of the slip-tube joints. A greater number of this type joint is in use due to its broad scope of application They are especially suited to high pressures and large pipes, but they are extensively and successfully used for low pressures and small pipes.

This type of joint is made with both external and internal guides as the space requirements permit. There seems to be no big advantage to either type of guiding over the other except that the internal guided joint occupies somewhat less space. As mentioned before it is very important that pipe lines be securely anchored between each joint so that the expansion for the entire line will be equally distributed among the individual joints and that no transverse stresses will be thrust into them. Typical anchor details are given in Figures 3 and 4.

The problems of packing for expansion joints are entirely different from those encountered in the design of stuffing boxes for valves, pistons, and other rapidly moving parts. Packing in an expansion joint fails in a way analogous to the formation of coal. Coal forms under extreme heat and pressure due to which it undergoes a process of solidification. Improper packing, as in the case of coal, hardens under the temperature of the pipe line and is compressed into a solid mass by the pressures of the stuffing glands. The packing then looses its resilience and the improper lubricants are distilled off under operating temperatures so that it becomes a brittle mass. Through careful study and research the above weaknesses have been overcome in

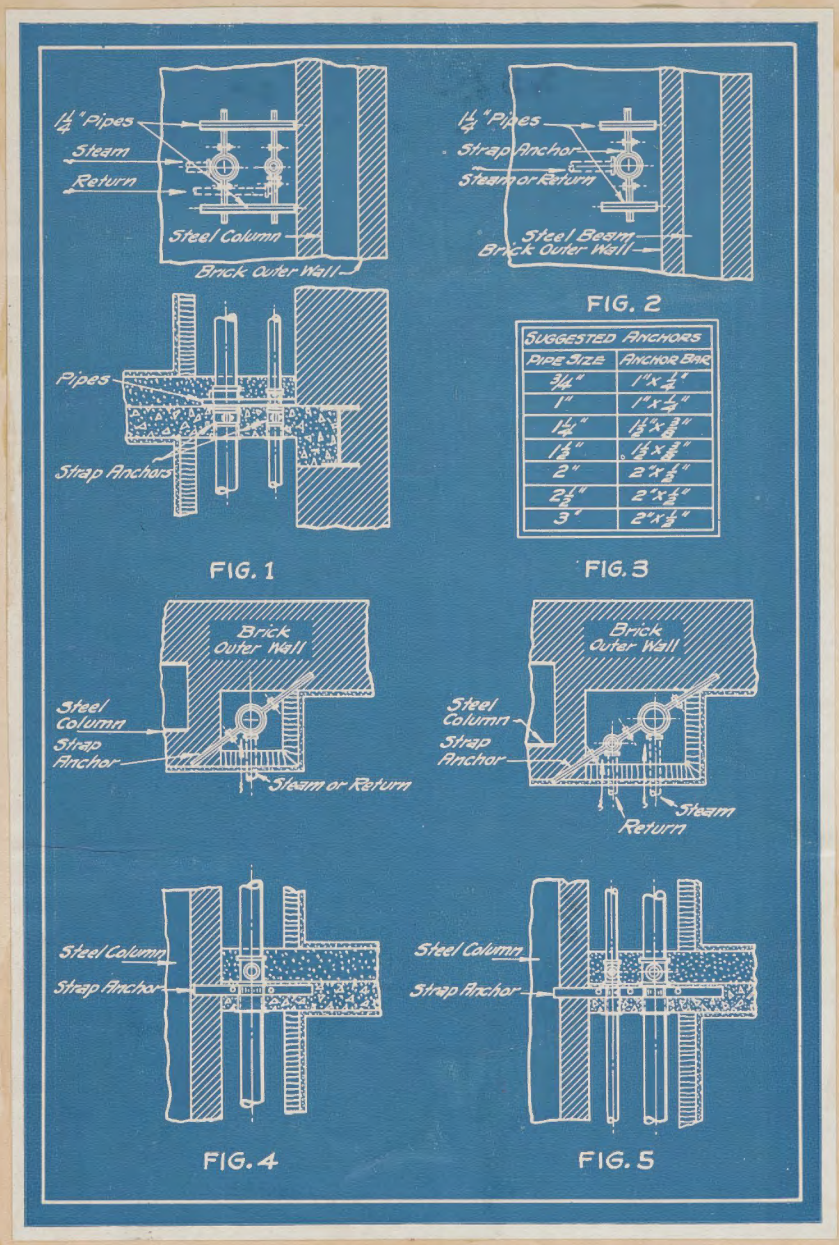

Figure 3. Anchor Details For Steam Risers.

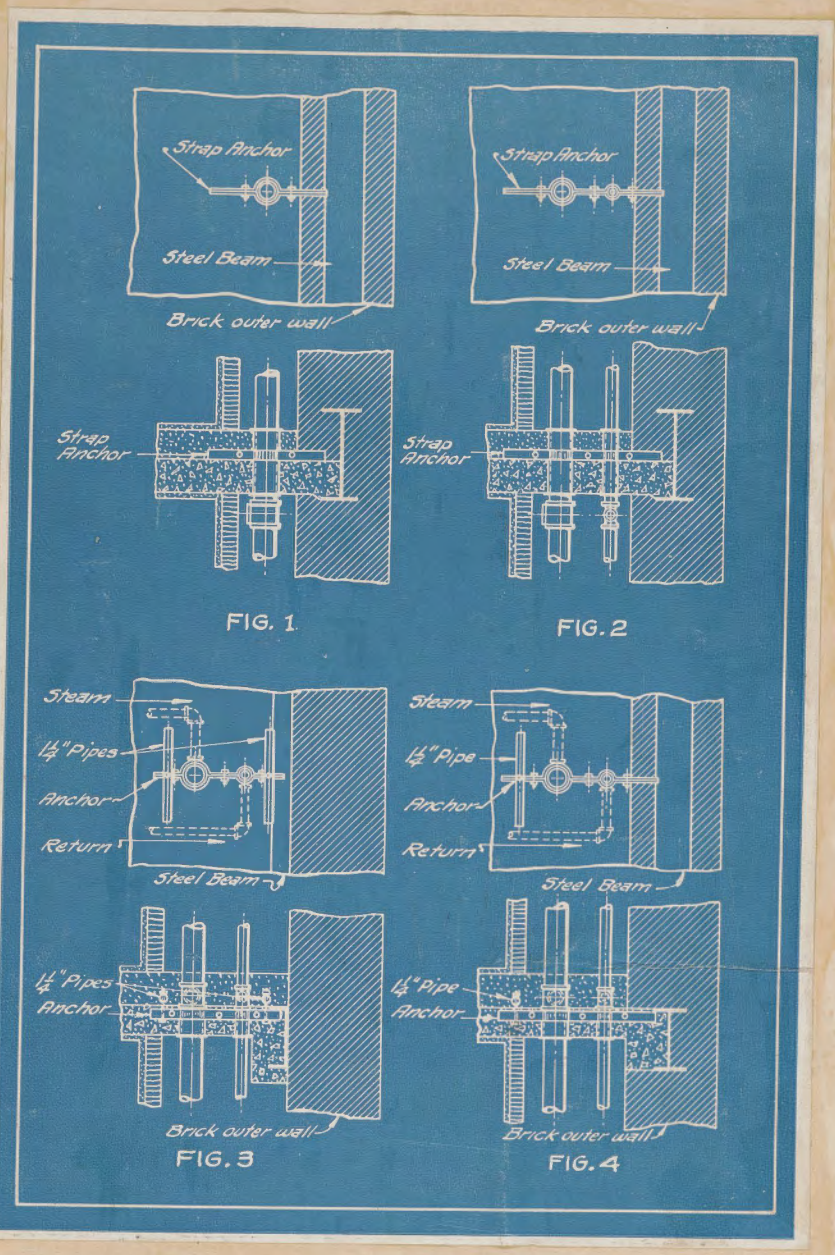

Figure 4. Anchor Details For Steam Risers.

the modern packings. Pictures of the two kinds of slip-tube joints are given in Figures 5 and 6.

The packless or bellows type of expansion joint incorporates the general principles of the slip-tube joint in that there is definite provision for maintaining the alignment of the shell and the slip-tube through guides or slots. In place of the stuffing box are a series of deep bellows that expand or contract with the movement of the pipes.

This type of expansion joint has a definite field of application, being best suited to low pressure work. Expansion joints of the bellows type are designed for a short traverse and with such depth of corrugations that the metal is not subjected to stresses at any point that exceed ten or twenty per cent of its elastic limit. Although it was first thought that corrugated packless joints were good for pressures up to 25 lbs. per square inch only, they are now being used successfully for as high as 200 lbs. per square inch pressures. They are made as a single unit from seamless copper tube, properly designed and rolled, and fitted with cast iron or steel equalizing rings to equally distribute the expansion on each corrugation. The principle advantage claimed by the packless bellows type of expansion joint is that when once installed no further attention is necessary.

A cut-out picture of one of the Sylphon packless joints made by The Fulton Sylphon Company is given in Figure 7. This is the type of joints on which tests were made for this paper. The methods used in determining the efficiency of practical

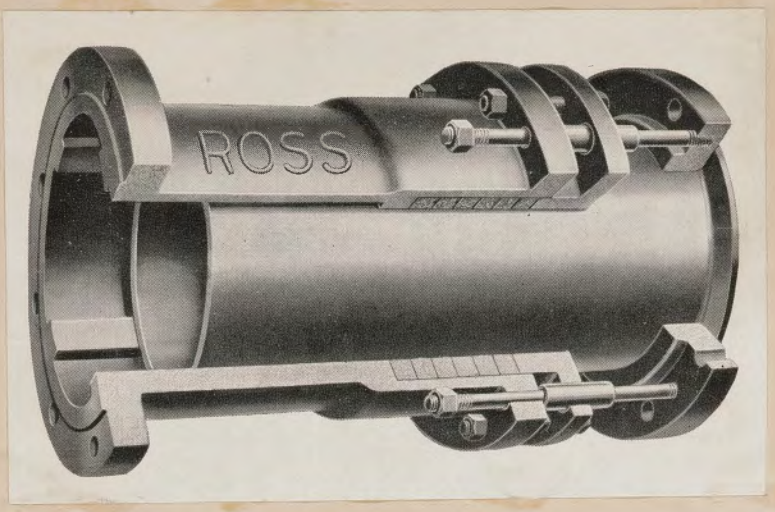

Figure 5. Ross Internally-guided Joint

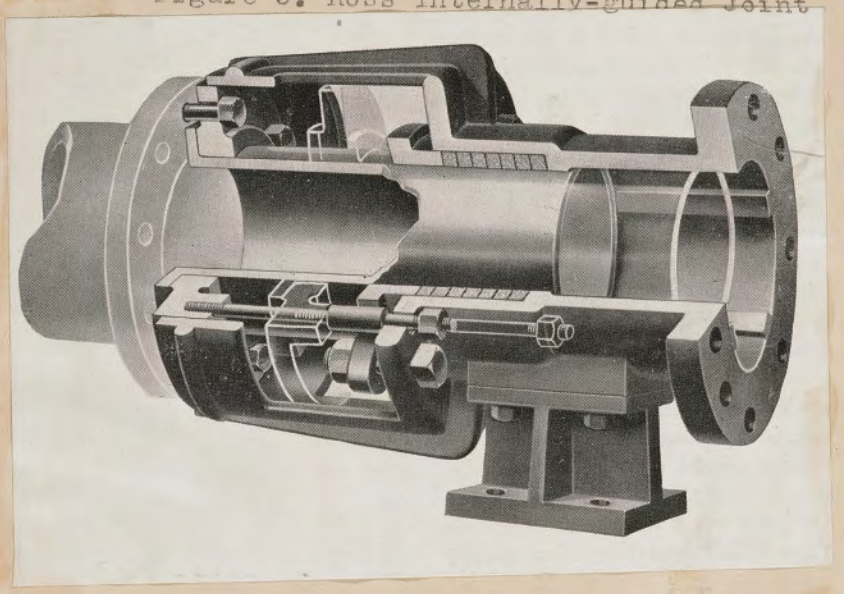

Figure 6. Externally-Guided Joint.

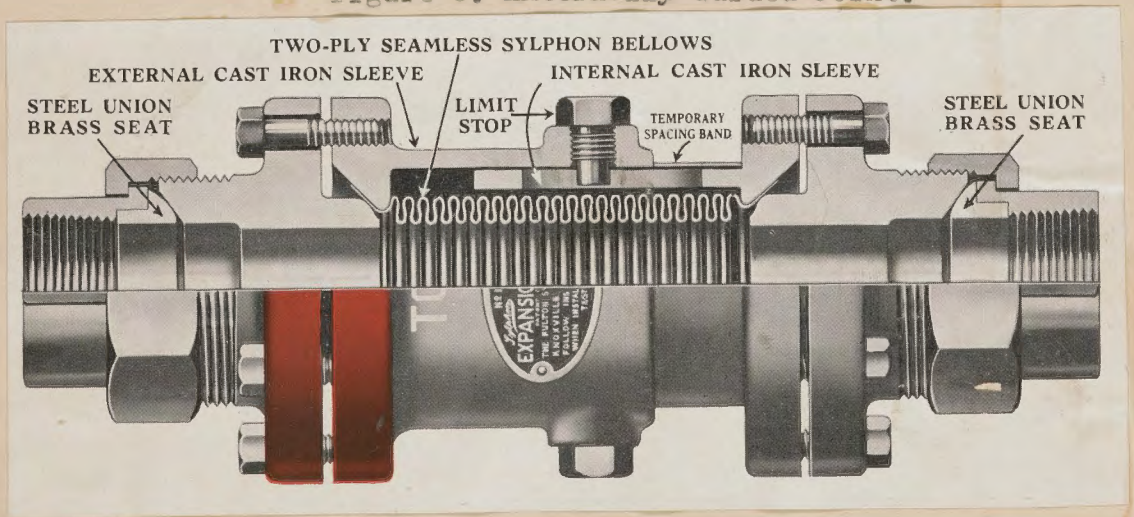

Figure 7. Sylphon Packless Expansion joint

operation of this type of joint were to test the endurance of the joint in as nearly as possible the working conditions experienced on the job, but speeded up to obtain results more quickly,and to determine the average number of strokes they make per day on the real job. With this data the length of time that they would last in service, if they failed wholly due to fatigue, can be calculated. If they do not last very nearly this long it is evident that there are other factors to be considered in figuring the life of the joints in service. The other factors are searched for by exposing the bellows to steam containing various kinds of boiler compounds. After they have remained in these compounds for a period of time they are re moved and given an endurance test. With the lives of these bellows before one it is possible to find the compounds that are the most injurious to the metal.

Sensitivity of operation is determined by giving a static test to the bellows, finding the dead weight required to make all strokes from zero to a maximum. Also pressure tests were made to determine the amount of pressure that it is necessary to apply in the form of steam to cause the assembled joint to make the full stroke. Another pressure test was made to see if the joints would withstand the pressures that they were likely to encounter in service. This was done with water pressure since the highest pressure they are likely to meet with is the weight of the columns of condensate trapped in the return lines.

DESCRIPTION OF THE APPARATUS AND METHODS USED

In order to determine the efficiency of the operation of the Sylphon packless expansion joint in service, it was necessary to design and set up a machine that would, as nearly as possible, duplicate service conditions. The accompanying photograph in Figure 8 shows the general set-up and arrangement of the apparatus for this test. The photograph does not show the pressure regulator which is installed in the steam supply line near the ceiling nor the exhaust line which lies near the floor and carries the exhaust steam from the room. This machine was designed and built especially for this job. It is designed so as to subject the joints to a most severe test. Four joints, of any size up to and including the 2 inch, may be tested on this machine at the same time. Less than four may be tested, if desired, as each position is equipped with an individual control valve. When testing joints larger than 2 inches, it is necessary to have two positions cut out as the machine has not the space for installation of more than two of the larger sizes at the same time. It will be noticed that the joints are tested in a horizontal position, which increases the severity of the test. The machine operates at a speed of 100 R.P.M., thus causing the joints to make 100 complete strokes per minute.

The machine is equipped with cams or eccentrics which have a $1\frac{1}{4}$ inch throw. This is equivalent to the maximun stroke for which the joints are designed. These eccentrics are set 90 degrees apart so that the joints will complete their stroke in the order 1-4-3-2, taking 1 as being the front position in the photograph. The motion of these eccentrics is transferred

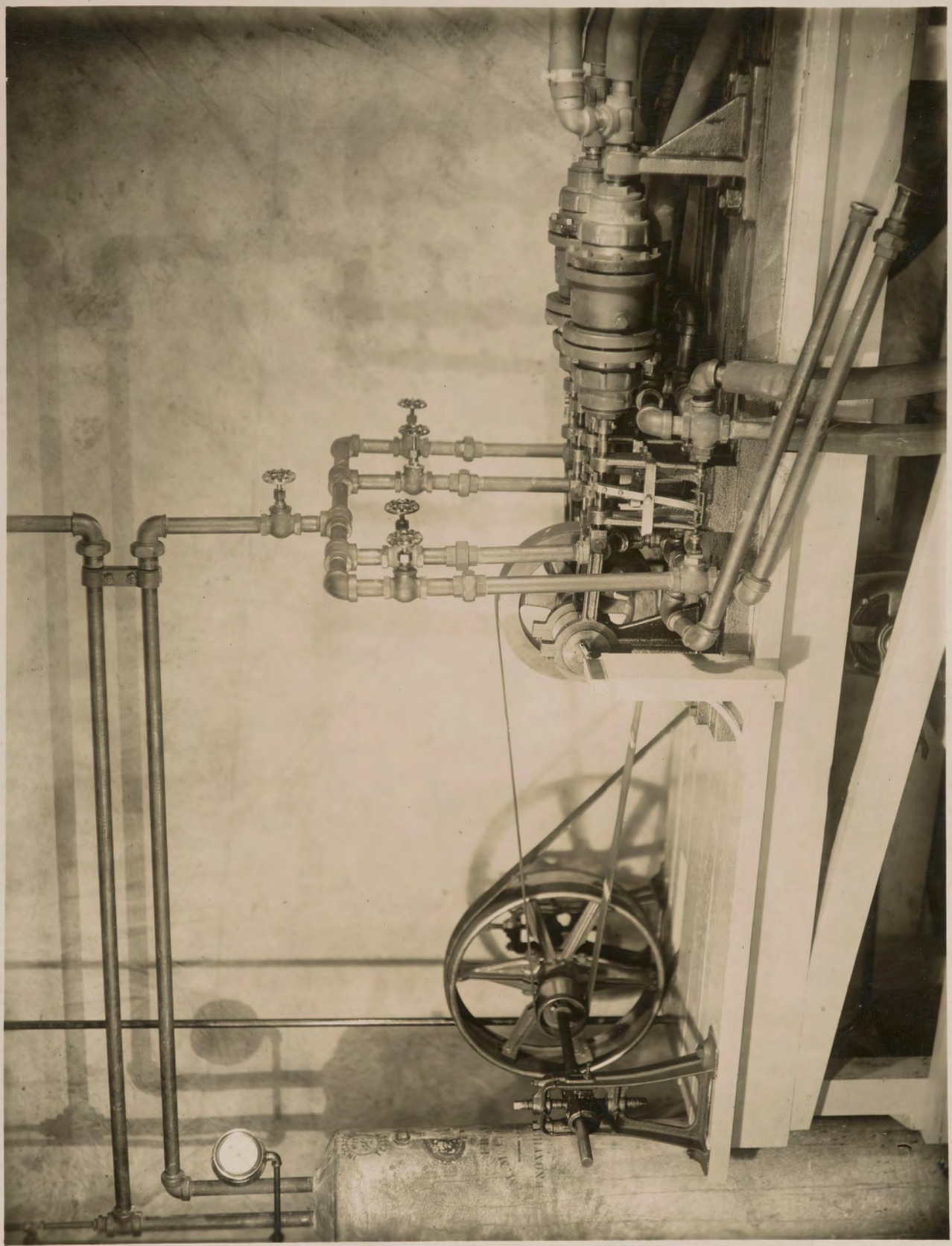

Figure 8. Testing Machine

to the joints by means of a connecting rod and slide. This slide is supported by two uprights. Between these uprights, and fastened to the slide, is a pin which operates the rocker, which in turn operates the supply and exhaust valves. This is clearly shown in the picture. It will be seen that near the base of and connected to these slide supports by means of brackets, are two single seated disc type valves with their stems operating approximately in the same longitudinal line. These stems are fastened together by a bracket. The end of the rocker works in this bracket moving both stems simultaneously, thus opening one valve and closing the other. Each position is thus equipped. The one opened when the joint expands is known as the admission valve and allows steam to pass through a steam hose to the tail stock to which the joint is connected. In this way steam is admitted to the joint as it expands and the pressure gradually increases until the joint reaches its maximum stroke, at which time the pressure is a maximum. As the eccentric swings in the opposite direction the slide also changes direction, closing the admission and opening the exhaust valve. This releases the steam to the atmosphere allowing the pressure to decrease as the joint contracts. This is a duplicate of service operations.

A rotary counter is connected directly to the drive shaft of the machine. The life of a joint is determined by the difference in the counter reading at the time it is started and the time when it breaks. The machine is driven by a 3 H. P., 220 volt, 3 phase motor which is mounted on a shelf under a table directly below the machine. The steam supply is taken from the

laboratory supply line and reduced to 5 pounds per square inch gage pressure with a Sylphon pressure regulator. It then passes into the large tank, seen at the left of the photograph, which acts as a reservoir and separator. The machine draws its supply from this tank. This insures a large, well-regulated supply of steam throughout the test.

After the machine is set for any particular size joint, the procedure used for installing the joint is the same as that used for installation in a pipe line. All joints were installed in a similar position, with the end marked "top" attached to tail stock and the name plate, which is placed directly below this marking, faced upward. This end, being fastened to the tail stock, naturally became the stationary end. Particular care was taken in the alignment of the joints to insure that they moved in a straight line.

Sylphon packless expansion joints are made in the following sizes: 1, 1¼, 1½, 2, 2½, and 3 inches. The 1 and 1¼ inch joints contain 2 inch bellows, the 1½ inch joints contain 2 7/16 inch bellows, the 2 inch joint contains 3 inch bellows, the 2½ inch joints contain 3 5/8 inch bellows, and the 3 inch joint contains 4 inch bellows. Joints containing bellows of all of the above sizes were tested. The following thicknesses of bellows were used in testing each of the above joints: 0.006, 0.007, 0.008, and 0.009 inch. These bellows were made from 80:20 brass metal.

Also bellows were tested of the 80:20 brass composition in thicknesses of from 0.008 to 0.010 inch for each size. Joints containing all of the above sizes of bellows made from a metal called "everdur", which is a trade name given to that

particular composition of brass, were tested. The minimum, maximum, and average life, the number of each size and kind tested, and the date tested are given in tables 1 to 5 of this paper under the heading of "Results." It will be noticed from these tables that all of the joints were not run until the bellows broke. They were removed unbroken because it was found from tests on the real job that if a joint would last 50,000 strokes, it would last the average life time of a building.

From tests made on jobs in six multi-story buildings in New York, it was found that the average strokes per day, over a two month period, were two and the average length of strokes was about 1 inch which is only two-thirds of the total stroke for which the joints are designed.

Two joints containing each of the different thicknesses of bellows were given pressure tests on the machine described above. The only difference in the set up is that water pipes are connected to the tail pipe of the joints on the machine. To test the joints, first 50 pounds per square inch pressure is let on one of the two joints of each kind. Then the bellows is removed and the effect noticed. If the bellows is still in good condition the other joint is esposed to 75 pounds per square inch pressure. The pressure was controlled by a hand valve. This bellows is removed and the effect of the pressure is noted. This is the extent of the pressure test because the joints are designed for only 25 pounds per square inch pressure and if the bellows withstands either the 50 or the 75 pounds

Per square inch pressure, they will undoubtedly withstand service pressures. The greatest pressures that the joints are likely to experience in operation are columns of condensate which may be blocked in the steam lines due to failure of suction pumps of cut-off of return lines. Fifty pounds per square inch in pressure is equivalent to 115 feet of water which is greater than the depth of condensate is likely to be. The machine was not running while this test was made.

One tube of each of the thicknesses mentioned above was given a static pressure movement test to determine the sensitivity of each of them. This will give a very good sensitivity test for the joints, themselves since friction in the joints was found to be almost neglible when in perfect alignment. In order to determine the stresses in the bellows the dimensions of the bellows must be determined accurately. By means of a dial gage and a laboratory rod the depth of the corrugations was measured. The inside diameters were found by subtracting twice this depth of corrugations from the outside diameters. The thickness of bellows was taken at both inner and outer bends of the corrugations with micrometer calipers. The static properties were taken by means of a static machine which is composed of a pair of balances set within a frame that is so built that weight can be applied to the bellows and the amount of weight registered by the scales. The deflection of the bellows is registered by an attached dial guage. Readings were taken at intervals of two pounds except in case of the 0.009 inch bellows on which five pound intervals were used. From

these data the static or load deformation curves were plotted, and from the curves the loads required to compress the bellows 1¼ inches were determined. With the necessary data now at hand, the range of stresses for each bellows was calculated. The data is included in table 6 for all of the above tests as to static properties. The load-deformation curves are given in figure 9. Since the curves are given, the data from which they were plotted is not included. The calculations of the stresses will not be shown because the formulae that was used cannot be exposed since it is the only formula of its kind known to exist. The Fulton Sylphon Company asked that it not be used. It is used very successfully in estimating the life of a bellows which is to be subjected to a certain length of stroke without testing by getting the highest stresses due to pressure and bending at the inner and outer corrugations and comparing them with bellows of like stresses that have been tested heretofore.

The stroke of 1¼ inch for which the joints are designed will take care of about eighty feet of pipe if the temperature change is not more than will be brought about by a change in pressure of about 50 pounds per square inch. A sketch of the arrangements of these joints in the pipe lines of a high building is given in figure 10.

The physical weight of forty-four of the bellows that were tested was found and comparison of the weights made and found that the greater the variation in weight the greater the variation in life for bellows of the same thickness. Their

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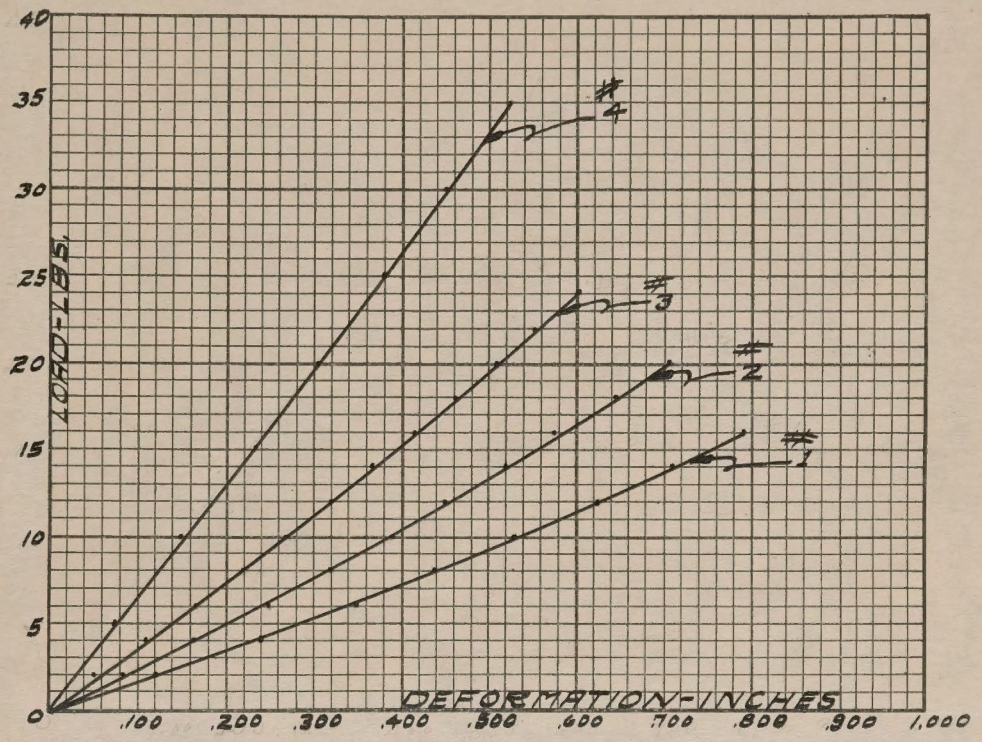

FIG. NO. 9.

TITLE:-

LOAD-DEFORMATION CURVES FOR EXPANSION JOINT BELLOWS MADE FROM TUBES OF DIFFERENT THICKNESSES.

INTERPRETATION OF CURVES

CURVE NO.1 - BELLOWS MADE FROM TUBE 0.006" THICK.
CURVE NO.2 - " " " " 0.007" " .
CURVE NO.3 - " " " " 0.008" " .
CURVE NO.4 - " " " " 0.009 " .

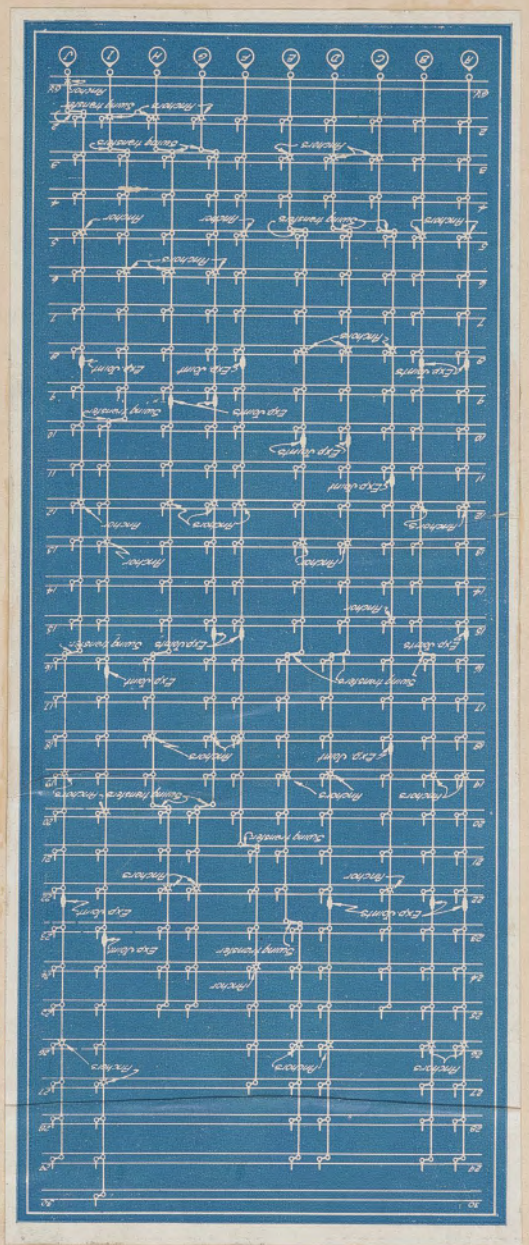

Figure 10. Arrangement of Expansion Joints in A High Building

weights are given in table 7. Since some of the joints have failed in service due to break in bellows and it is found from experiment that they will withstand more full strokes than they have made in service, there must be some other weakness of the bellows other than bending and pressure stresses. It was found by exposing these bellows to solutions containing only small traces of ammonia, that they would lose a certain part of their elasticity and therefore break much sooner. In analysis of many of the steams to which these joints are exposed, both chlorine and ammonia traces have been found which to some degree accounts for the shorter life of the bellows than was anticipated.

A one inch joint manufactured by the Ross Heater Company of the packless corrugated type was tested in comparison with a one inch joint manufactured by the Fulton Sylphon Company and the former had a life of nearly four times the latter. The Ross joint was much harder to keep in alignment and as a result the guides wore very badly. It made 97,860 strokes before it failed and the Fulton joint made only 25, 502 strokes.

RESULTS

TABLE NO. 1

1" & 1-1/4" JOINTS, 2" BELLOWS

Metal	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	Everdur
Date Tested	May 1928	June 1929	July 1930	July 1930	July 1930	July 1930	August 1931
Tube Thickness	.008-.010"	.008-.010"	.006"	.007"	.008"	.009"	.006-.007"
No. Tested	9	4	7	14	12	11	22
Min. Life	28,320	9,173	23,790	5,183	20,786	17,360	42,900
Max. Life	56,270	18,900	112.030	120,811	134,560	28,520	379,200
Avg. Life	38,916	14,420	64,081	54,615	53,118	22,601	169,500

TABLE NO. 2

1-1/2" JOINTS, 2-7/16" BELLOWS

Metal	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	Everdur
Date Tested	June 1928	JUNE 1929	Nov. 1930	Nov. 1930	Nov. 1930	Nov. 1930	August 1931
Tube Thickness	.008-.010"	.008-.010"	.006"	.007"	.008"	.009"	.006-.007"
No. Tested	10	2	7	6	13	10	4
Min. Life	10,330	9,424	102,073	14,479	11,471	14,442	35,000
Max. Life	61,260	76,018	233,241	94,472	65,985	60,892	130,200
Avg. Life	31,604	42,721	153,899	66,516	41,330	28,769	83,750

TABLE NO. 3

2" JOINTS, 3" BELLOWS

Metal -	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	Everdur
Date Tested	June 1928	June 1929	Nov. 1930	Nov. 1930	Nov. 1930	Nov. 1930	February 1932
Tube Thickness	.008-.010"	.008-.010"	.006"	.007"	.008"	.009"	.006-.007
No. Tested	10	2	13	8	15	8	4
Min. Life	21,120	36,690	32,685	51,909	17,351	12,392	75,600
Max. Life	48,773	40,608	208,766	90,479	60,081	27,560	100,000
Avg. Life	36,222	39,649	111,033	74,563	39,484	17,418	90,050

TABLE NO. 4

2-1/2" JOINTS, 3-5/8" BELLOWS

Metal	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	Everdur
Date Tested	June 1928	June 1929	Jan. 1931	Dec. 1930	Jan. 1931	Jan. 1931	March 1932
Tube Thickness	.008-.010"	.008-.010"	.006" (A)	.007" (A)	.008" (A)	.009" (A)	.006-.007"
No. Tested	13	2	5	4	6	5	4
Min. Life	36,334	133,489	50,000	50,000	50,000	50,000	230,000
Max. Life	98,790	195,800	50,000	50,000	50,000	50,000	330,800
Avg. Life	62,689	164,820	50,000	50,000	50,000	50,000	278,600

(A) These joints were run 50,000 or more strokes and removed unbroken.

TABLE NO. 5

3" JOINTS,2" BELLOWS

Metal	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	80:20 Brass	Everdur
Date Tested	May 1928	June 1929	Jan. 1931	Jan. 1931	Jan. 1931	Jan. 1931	April 1932
Tube Thickness	.008-.010"	.008-.010"	.006" (B)	.007" (B)	.008" (B)	.009" (B)	.006-.007"
No. Tested	2 (A)	2 (A)	5	8	8	13	4
Min. Life	100,000	151,732	50,000	50,000	50,000	50,00	368,000
Max. Life	100,000	243,230	50,000	50,000	50,000	50,000	989,000
Avg. Life	100,000	197,841	50,000	50,000	50,000	50,000	733,600

(A) These joints were run 100,000 or more strokes and removed unbroken.

(B) These joints were run 100,000 strokes or more and removed unbroken.

TABLE NO. 6

Static Properties of Bellows

No.	Diameter		Thickness		Range of Stress			Load-lbs.
	Inside	Outside	Inner	Outer	Bending	Pres.	Resultant	Stroke-1¼"
1.	1.373	2.035	.010"	.009"	98,200	4,060	102,260	20.5
2.	1.380	2.036	.010"	.010"	146,800	3,985	150,785	31.0
3.	1.383	2.035	.012"	.012"	153,000	2,750	155,750	46.5
4.	1.393	2.043	.016"	.015"	166,000	1,660	167,660	82.0

TABLE NO. 7

EXPANSION JOINT BELLOWS INVESTIGATION

Reference No.	Tube Thickness	Break Corrugation	Break End	Break Quadrant	Weight-Pound	Life-Strokes
1.	0.006"	9th inner	Bottom	4th	0.346	23,790
2.	0.006"	6th inner	Bottom	2nd	0.337	77,879
3.	0.006"	3rd inner	Bottom	1st& 4th	0.346	43,192
4.	0.006"	4th inner	Bottom	3rd	0.328	112,030
5.	0.006"	2nd inner	Top	2nd& 3rd	0.345	77,110
6.	0.006"	3rd inner	Top	1st& 4th	0.347	90,700
7.	0.006"	5th inner	Top	1st	0.354	23,870
Avg.	0.006"	4.6 inner	4(B),3(T)		0.343	64,081
8.	0.007"	7th inner	Top	3rd	0.388	75,100
9.	0.007"	2nd inner	Bottom	2nd	0.373	28,211
10.	0.007"	6th inner	Top	1st	0.387	79,980
11.	0.007"	4th inner	Bottom	2nd	0.390	42,970
12.	0.007"	5th inner	Bottom	3rd	0.389	120,811
13.	0.007"	4th inner	Bottom	1st	0.387	40,670
14.	0.007"	4th inner	Top	3rd	0.386	47,180
15.	0.007"	8th inner	Top	4th	0.376	113,200
16.	0.007"	4th inner	Top	4th	0.387	5,183
17.	0.007"	5th inner	Top	1st	0.410	31,460
18.	0.007"	2nd inner	Bottom	3rd& 4th	0.389	44,440
19.	0.007"	2nd inner	Top	1st&2nd	0.390	43,400
20.	0.007"	4th inner	Bottom	1st	0.387	58,000
21.	0.007"	4th inner	Top	3rd& 4th	0.392	34,000
Avg.	0.007"	4.4 inner	6(B),8(T)		0.388	54,615
22.	0.008"	2nd inner	Top	3rd& 4th	0.426	20,786
23.	0.008"	10th inner	Bottom	4th	0.420	46,490
24.	0.008"	2nd inner	Top	3rd	0.428	30,690
25.	0.008"	4th inner	Bottom	3rd	0.429	34,630
26.	0.008"	2nd inner	Bottom	3rd& 4th	0.427	33,000
27.	0.008"	3rd inner	Bottom	1st	0.425	84,500
28.	0.008"	2nd inner	Bottom	1st	0.427	32,000
29.	0.008"	2nd inner	Bottom	2nd& 3rd	0.426	66,760
30.	0.008"	4th inner	Top	1st&4th	0.425	55,710
31.	0.008"	5th inner	Bottom	1st	0.425	47,360
32.	0.008"	2nd inner	Bottom	2nd& 3rd	0.423	50,930
33.	0.008"	2nd inner	Top	4th	0.426	134,560
Avg.	0.008"	3.33inner	8(B),4(T)		0.426	53,118
34.	0.009"	2nd inner	Top	3rd	0.504	19,300
35.	0.009"	2nd inner	Top	2nd& 3rd	0.505	28,520
36.	0.009"	2nd inner	Bottom	2nd	0.504	21,483
37.	0.009"	1st outer	Top	2nd& 3rd	0.507	28,340
38.	0.009"	2nd inner	Top	1st& 2nd	0.503	22,630
39.	0.009"	2nd inner	Bottom	4th	0.506	21,000
40.	0.009"	2nd inner	Top	2nd& 3rd	0.505	17,360
41.	0.009"	1st outer	Top	4th	0.505	21,365
42.	0.009"	2nd inner	Top	2nd	0.504	19,260
43.	0.009"	1st inner	Top	3rd	0.504	21,900
44.	0.009"	2nd inner	Top	3rd& 4th	0.504	27,458
Avg.	0.009"	1.73inner	2(B),9(T)		0.504	22,601

CONCLUSIONS

1. Within practical limits, the thinner the tube from which the bellows is made, the longer is the life of the joint.
2. The thicker the bellows, the nearer the end the break will occur.
3. The thicker the bellows, the smaller is the variation in weight.
4. The greater the variation in weight, the greater is the variation in life for bellows in the same group.
5. Greater flexibility of bellows increases their life.
6. The pressure stress, even for the thinner bellows, is not great enough to cause much damage.
7. All of these joints will stand 75 pounds per square inch hydraulic pressure.
8. The movement is not causing serious damage to these joints in service.
9. The action of the steam and the boiler compounds within the steam have a serious effect on these bellows in service.

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APPROVAL

The foregoing thesis is hereby approved as a creditable study of an engineeting subject, carried out and presented in a manner sufficiently satisfactory to warrant its acceptance as a prerequisite to the degree for which it has been submitted. It is to be understood that by this approval the undersigned does not necessarily endorse or approve any statement made, opinions expressed, or conclusions drawn therein, but approves the thesis only for the purpose for which it is submitted.

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