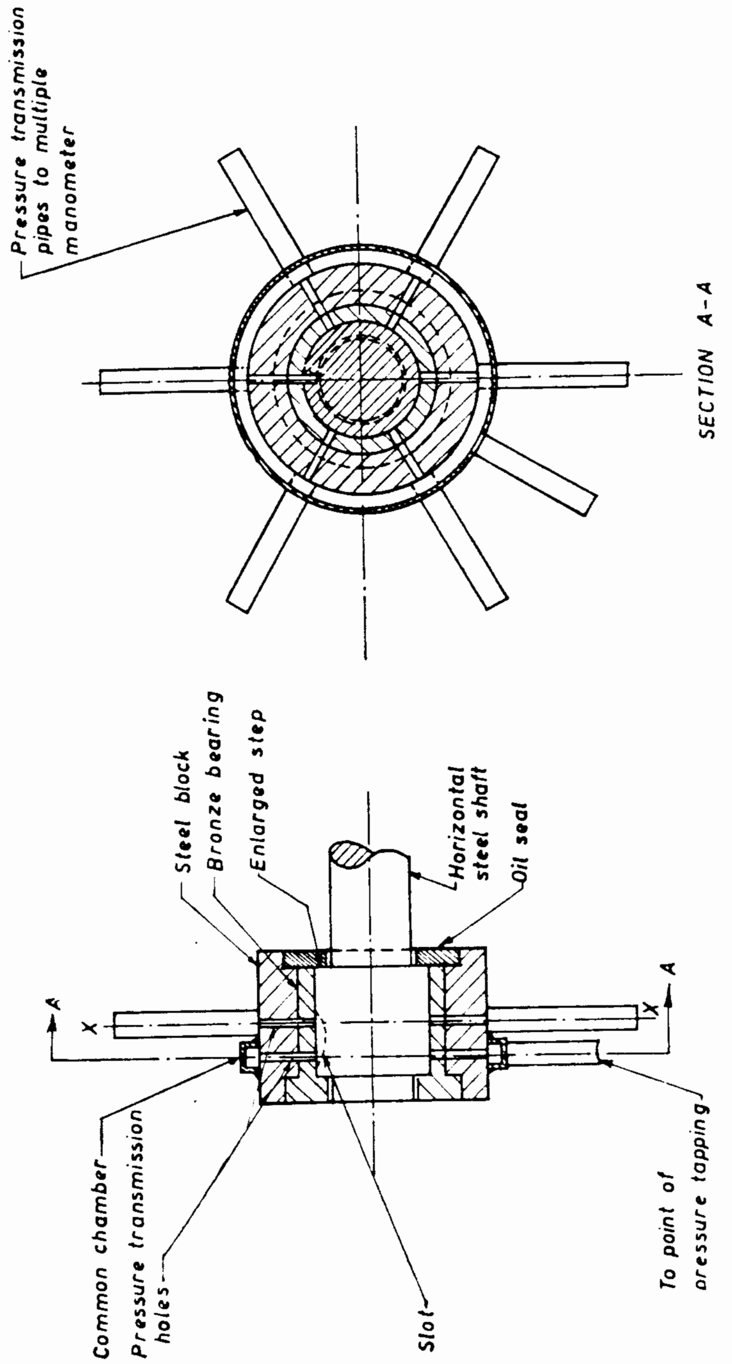

FIG. 8. Rotary Valve Indicator.

a series of manometers or other pressure measuring devices could give the pressures at any given point in the pipe at successive intervals of time.

The instrument essentially comprises a rotary valve and a sectioned drawing is given in Fig. 8. This consists of a horizontal steel shaft with an enlarged step, rotating in a bronze bearing housed in a steel block. Six pressure transmission holes, 1/16″ in diameter and equally spaced around the circumference are provided in each of the planes AA and XX in such a manner that each of the holes in one plane is exactly opposite to a corresponding hole in the second plane. Each hole in plane XX is connected to a separate manometer by the pressure transmission pipe while all the holes in plane AA lead to a common chamber which is connected to the point at which the pressure wave form has to be determined. A slot in the shape of a feather key-way is provided on the machined step in such a way that if the shaft is rotated corresponding holes in the two planes are connected at successive intervals of time.

Air-tightness was ensured by the oil film in the bearing since the oil-seals were sufficient for the pressures encountered. The shaft of the rotary valve was directly connected to the shaft operating the cam.

The rotary valve was connected to a multiple manometer (Fig. 3 and Fig. 9) in the manner commonly used in wind tunnel testing.[24] The manometer tubes were filled with water coloured blue with methylene dye and a few drops of photographic wetting agent were added to lower the surface tension.

The readings obtained on the multiple manometer with the help of the rotary valve mechanism were not representative of the actual static pressure in the pipe at the point of the pressure tapping due to the interaction of the intervening air space.* Pressures measured at the entrance of the heat transfer section, however, would allow to correlate the heat transfer measurements with the steepness of the differently shaped wave forms (see Fig. 10).

5. EXPERIMENTAL TECHNIQUE

The experimental set-up was designed to allow the determination of the total quantity of heat transferred through the wall of a horizontal tube to pulsating air from the surrounding steam jacket under different conditions of air mass flow, frequency of pulsations, pressure amplitude and wave form. The major quantities measured were therefore air and condensate flows, the temperature of the steam in the jacket and of the air before and after heating and the frequencies of the pressure waves, for different shapes and amplitudes.

First, preliminary experiments were conducted to ascertain the accuracy of and to calibrate, the various measuring arrangements that were used in the final experiments.

* Accurate measurements of pressure and velocity of oscillations of large amplitudes induced on the air in a long pipe by an oscillating piston have been undertaken by Lettau.[25]

The boiler was the first part of the apparatus to be switched on. The steadiness of the conditions in the heat exchanger was judged from the rate of condensate formation. As soon as steady temperature conditions were achieved, the valve imparting the pulsations was kept in the open position and compressed air was allowed to pass through the tube. The rate of flow was adjusted to the required value by regulating the entrance gate valve. When the temperatures of the air at inlet and outlet and the condensate formation became steady, readings were taken. From the measurements, the rate of heat transfer for steady flow conditions was calculated.

The motor to operate the valve was then started and set at its lowest speed. This changed the pressure difference across the orifice plate, which was restored to its original value by an adjustment of the entrance gate valve. When a steady state was again attained, readings were taken as under steady flow conditions. In addition, the motor speed was measured, which represented the frequency of pulsations.

The mass flow was kept constant and the rate of heat transfer was calculated for various frequencies. This was repeated at different mass flows for both the cams. Several sets of readings taken at different times showed very little discrepancy.

TABLE II

Heat Transfer with Steady Flow

Sl. No.	Re	Nu (Actual)	Nu (Theoretical)	Deviation %
1	9,210	24·16	30·72	– 21·4
2	10,310	27·24	33·65	– 19·0
3	11,770	30·01	37·41	– 19·8
4	13,620	33·62	41·98	– 19·9
5	14,280	35·18	43·65	– 19·4
6	14,800	35·54	44·87	– 20·8
7	16,090	38·52	47·97	– 19·7
8	17,890	42·10	52·24	– 19·4
9	19,640	45·12	56·36	– 19·9
10	21,500	48·15	60·53	– 20·5
11	22,380	49·75	62·52	– 20·4
12	23,450	51·65	64·86	– 20·4
13	26,330	54·64	71·12	– 23·2
14	27,820	58·70	74·30	– 21·0
15	29,400	61·85	77·80	– 20·5
16	30,410	63·30	79·80	– 20·7
			Mean Deviation %	– 20·4

6. EXPERIMENTAL RESULTS

Experimental data obtained in the manner described were evaluated and calculated as outlined in the Appendix. The following results were obtained:—

6.1. *Heat Transfer with Steady Smooth Flow.*

The data for steady smooth flow are evaluated and shown in Table II and plotted in Fig. 11. They may be represented by the equation:

$$Nu = \cdot 0191 \, Re^{\cdot 785}$$

In general these values lie an average of 20·4% below the Dittus-Boelter[26] line:

$$Nu = \cdot 0207 \, Re^{\cdot 8}$$

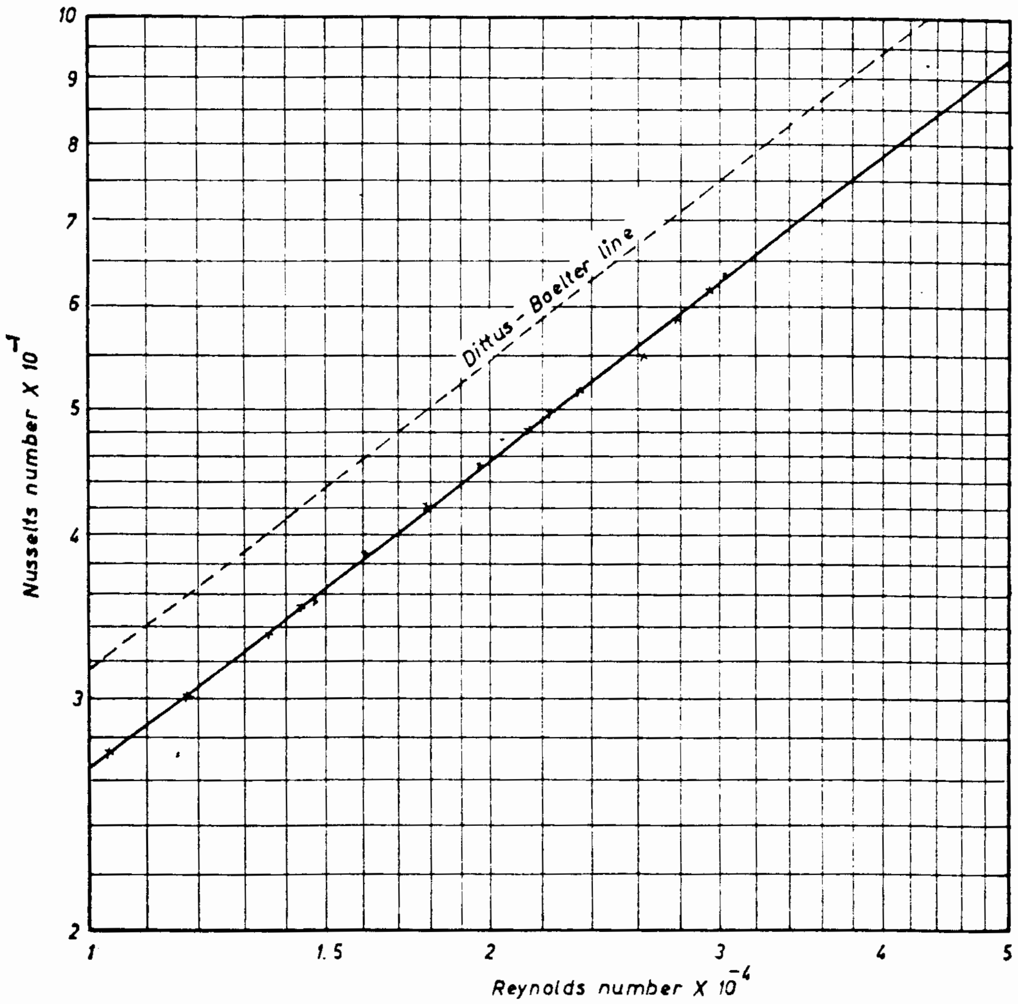

FIG. 11. Re *vs.* Nu for Steady Smooth Flow.

This is probably due to uneven heating of the heat transfer pipe along its length, the temperature being lower near the inlet end and higher at the outlet end. The tendency of these results, however, has no significance for the problem investigated here since the heat transfer coefficients with pulsating flow were compared with the coefficients of smooth flow under otherwise identical conditions.

6.2. *Heat Transfer with Pulsating Flow.*

The results obtained from the heat transfer experiments with pulsating flow are shown in Figs. 12 and 13.

With wave shape 'A', the ranges investigated were: $6 < n < 39$ and $6{,}400 < Re < 25{,}060$. The maximum reduction recorded, of the Nusselt number was about 43% at $n = 22 \cdot 5$ and $Re = 6{,}400$. The maximum increase was about 21% at $n = 35 \cdot 5$ and $Re = 20{,}060$. In general, the Nusselt numbers for Reynolds numbers $> 10{,}000$, were less than the steady state below a frequency of about 31 to 32 c/s. and greater above this value reaching the maximum at a frequency of about 36 c/s.

With wave shape 'B', the ranges investigated were $8 < n < 40$ and $6{,}400 < Re < 25{,}060$. The maximum decrease was about 20% at $n = 8$ and $Re = 14{,}970$. The maximum increase was about 14% at $n = 27$ and $Re = 6{,}400$.

With wave shape 'C', the ranges investigated were $9 < n < 26$ and $6{,}400 < Re < 14{,}970$. The Nusselt numbers were always less than those for steady flow, the minimum decrease being about 6% at $n = 15$ and $Re = 10{,}270$.

With wave shape 'D', the ranges investigated were $9 < n < 26$ and $6{,}400 < Re < 12{,}510$. The Nusselt numbers were always higher than those for steady flow within the ranges considered with a maximum improvement of about 42% at $n = 15 \cdot 5$ and $Re = 6{,}400$.

Fig. 12 shows typical curves of the Nusselt numbers against frequency obtained for all observed wave shapes and investigated Reynolds numbers. The Nusselt value for steady flow is also indicated in each figure. The same data are plotted in Fig. 13 in the form of the relative change of Nusselt number depending on frequency.

6.3. *Comments.*

It is seen from Figs. 12 and 13 that all experimentally obtained results do not fall on a smooth line though the trend of the interdependence is unmistakable. This emphasises the unsteady character of the process of heat transfer when it occurs in pulsating flow. A similar phenomenon has been noted by Schultz-Grunow[27] in his experiments on the velocity profile in pulsating water flow where the readings of the velocity at different points across the section of the pipe do not fall along a smooth curve. Yet the results obtained appear to be dependable in spite of the scatter since approximately the same curves were obtained by experiments repeated under the same conditions. The experiments with superimposed standing wave systems 'B' and 'D' show the same characteristics although in an accentuated manner.

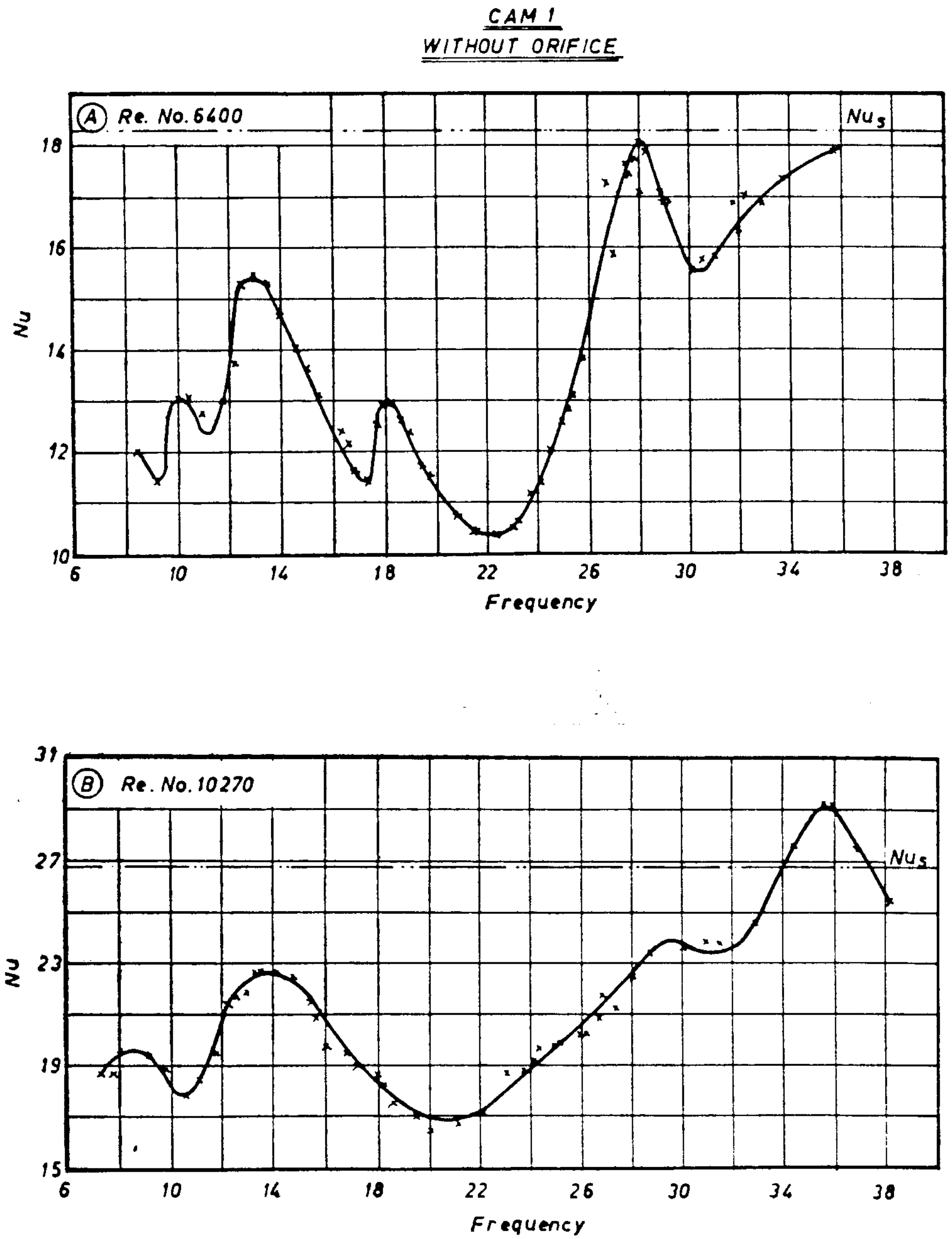

FIG. 12 (A, B)

6.4. *Conclusions.*

From the results of the experiments reported above it is possible to conclude that the process of h. .t transfer to a pulsating medium may be considerably modified

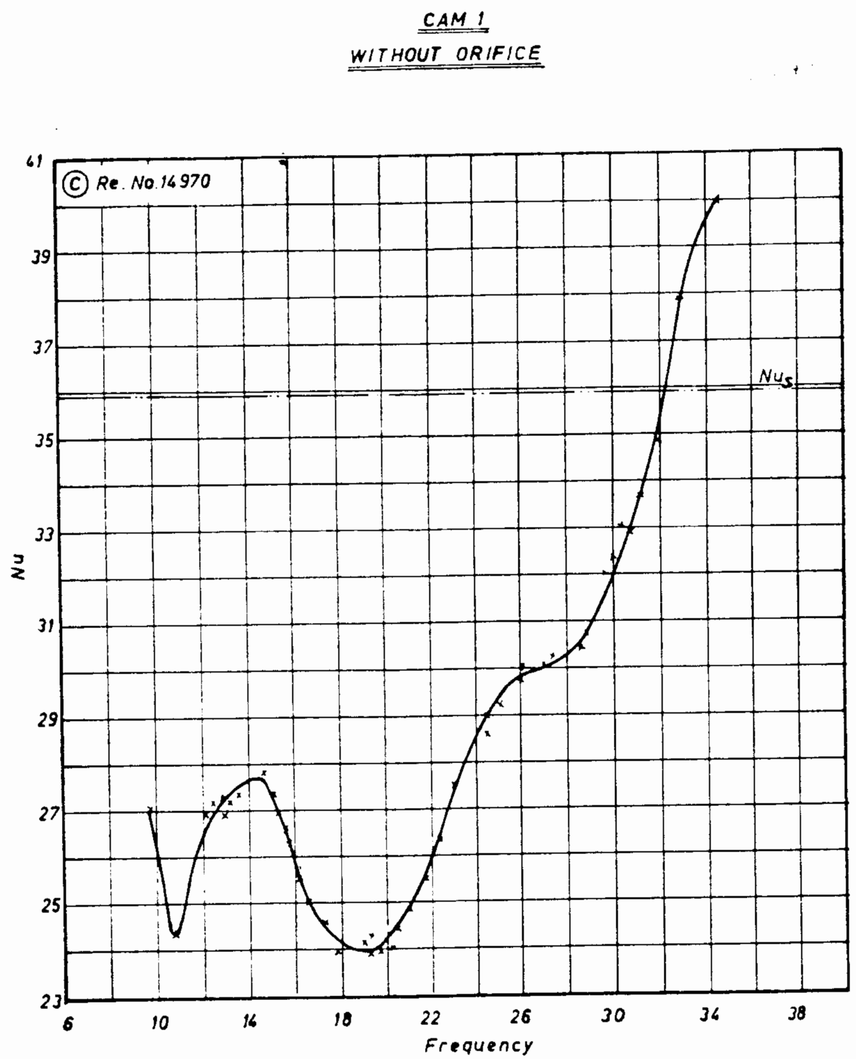

FIG. 12 (C)

by the nature and frequency of pulsations imposed on the medium. The change of the rate of heat transfer depends on the frequency of the pulsations and the Reynolds number whereby depending on them, negative or positive changes are encountered.

The onset of improved heat transfer occurs across a comparatively narrow band of frequencies and is maintained for a small range only for a cam which opens the passage for nearly half a revolution (see Fig. 13 A). For a cam opening only

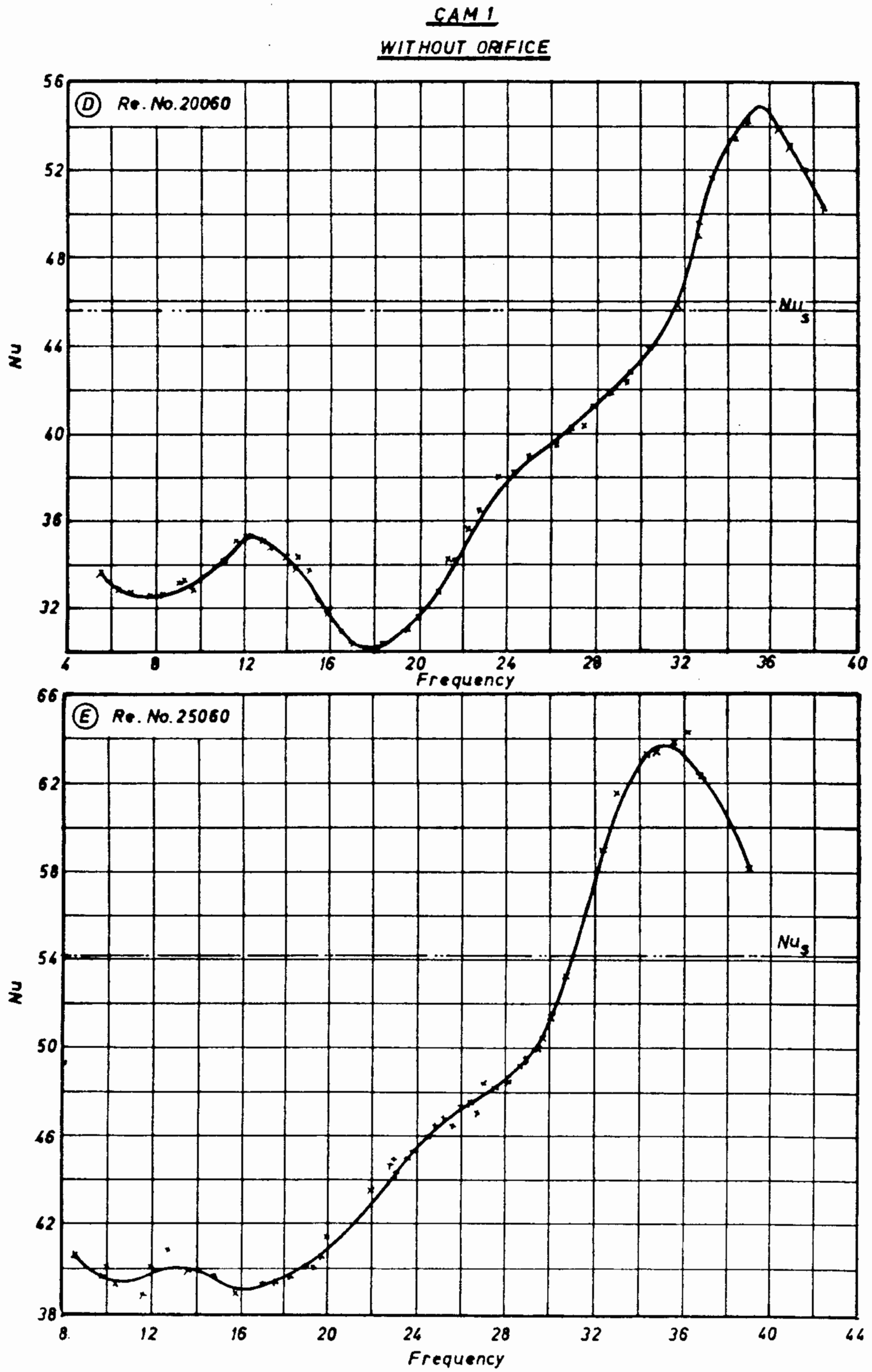

FIG. 12 (D, E)

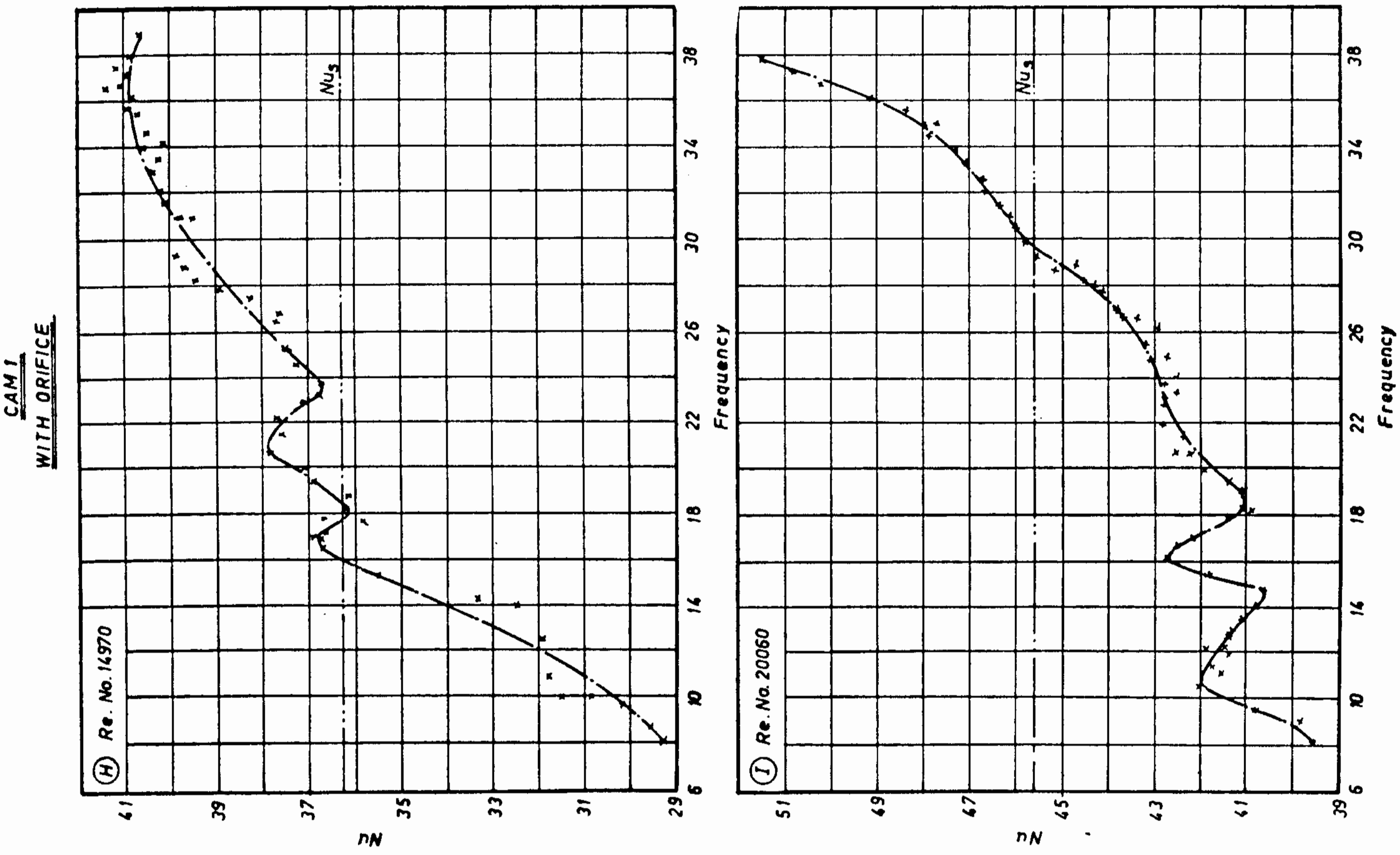

FIG. 12 (H, I)

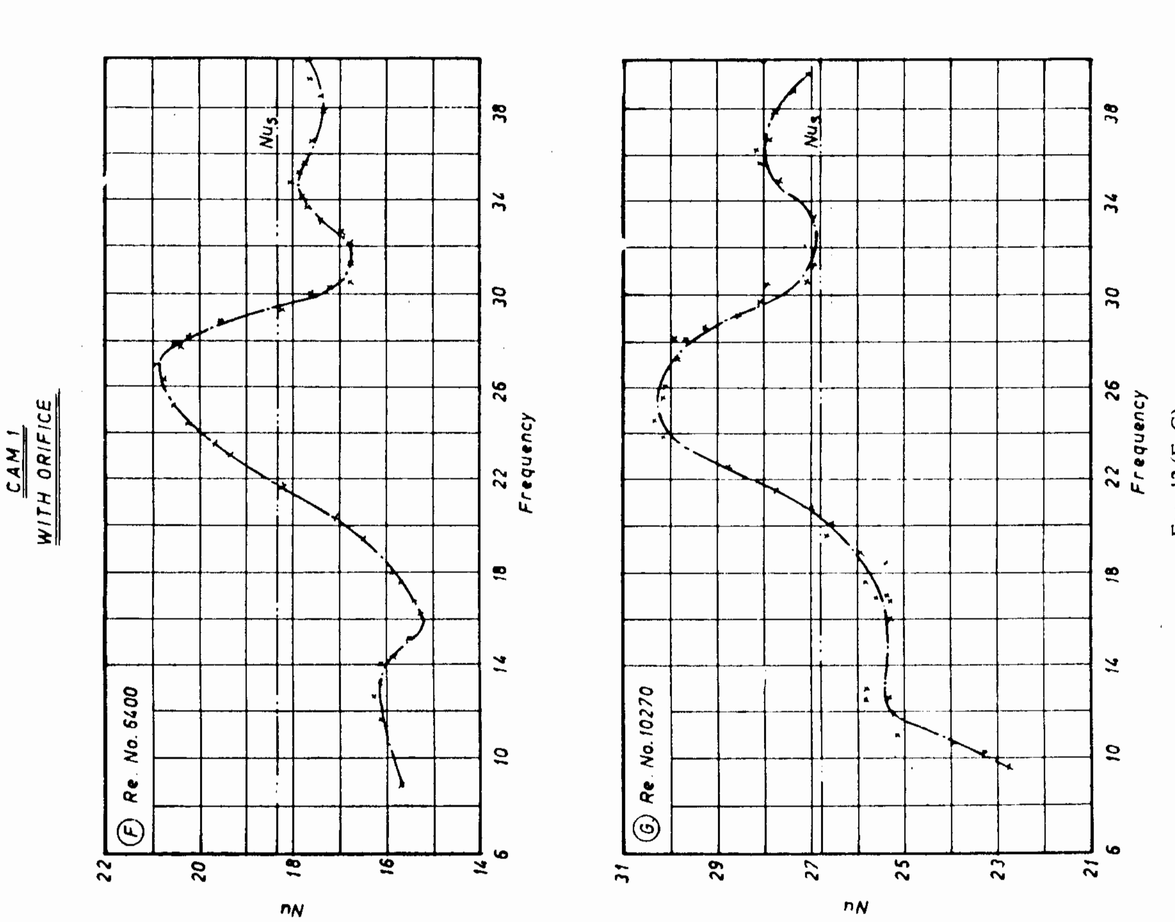

FIG. 12 (F, G)

CAM 2
WITHOUT ORIFICE

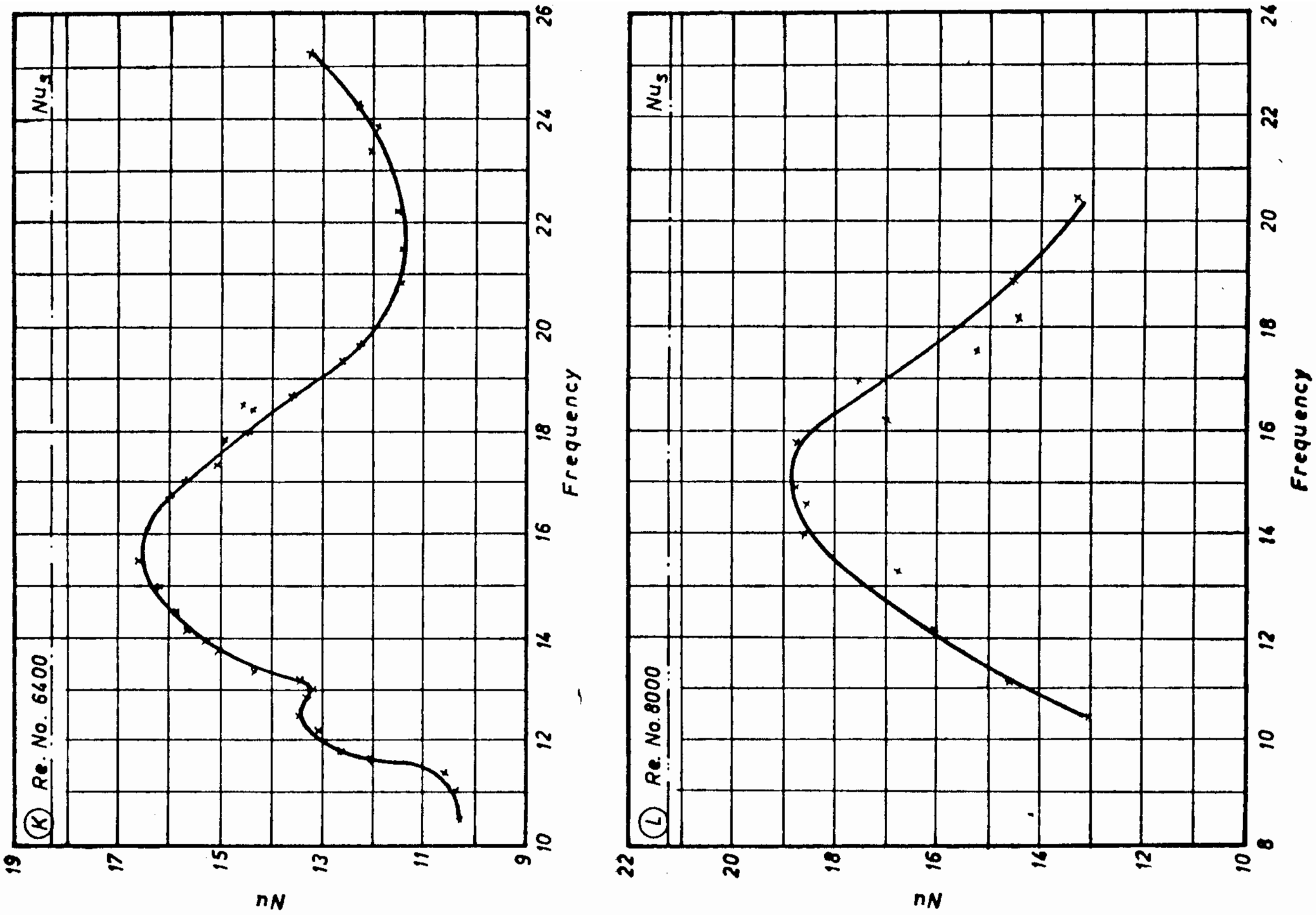

FIG. 12 (K, L)

CAM 1
WITH ORIFICE

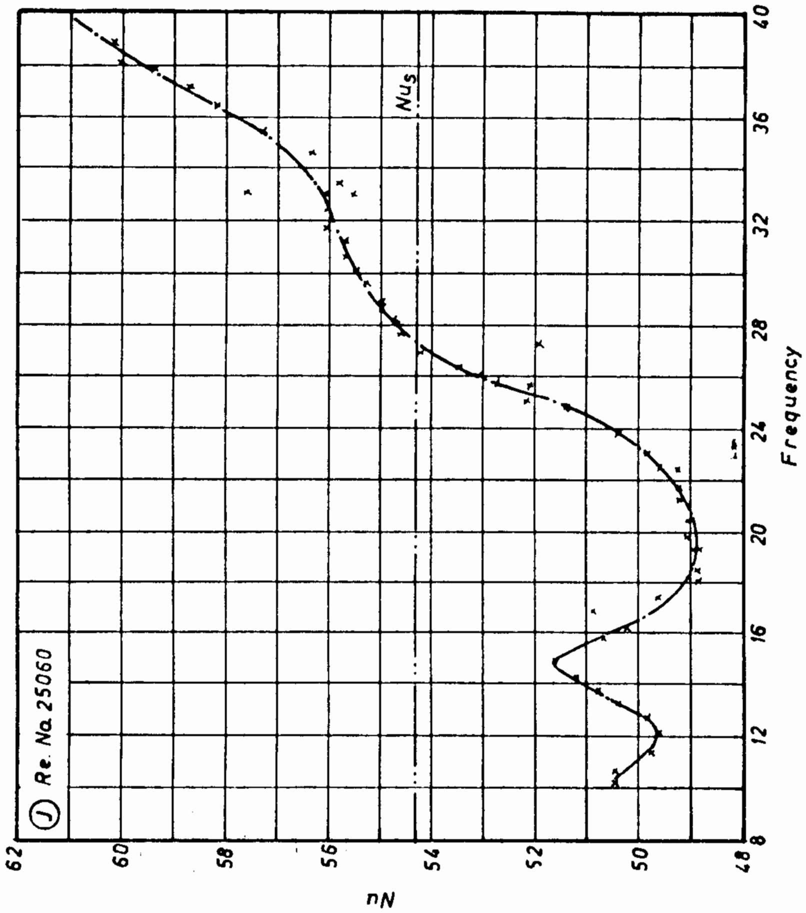

FIG. 12 (J)

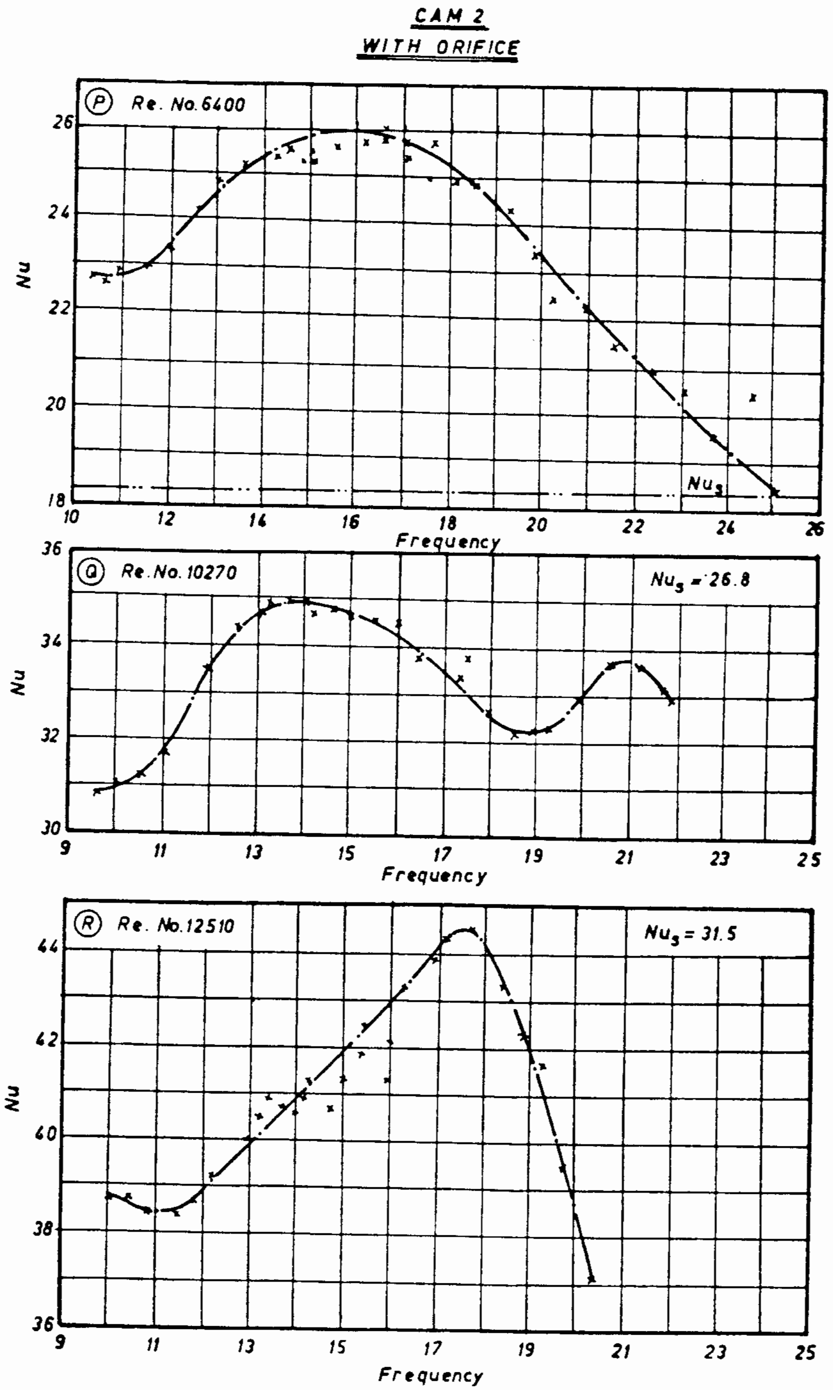

FIG. 12 (P, Q, R)

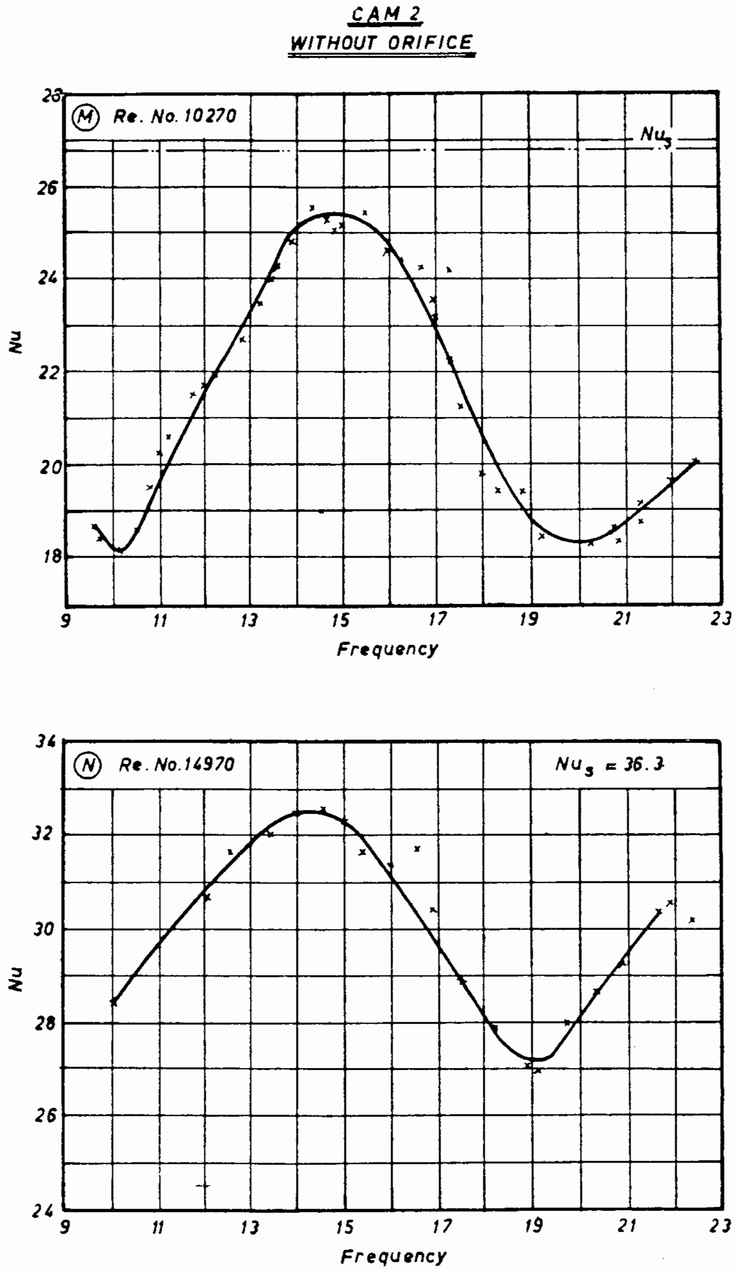

FIG. 12 (M, N)

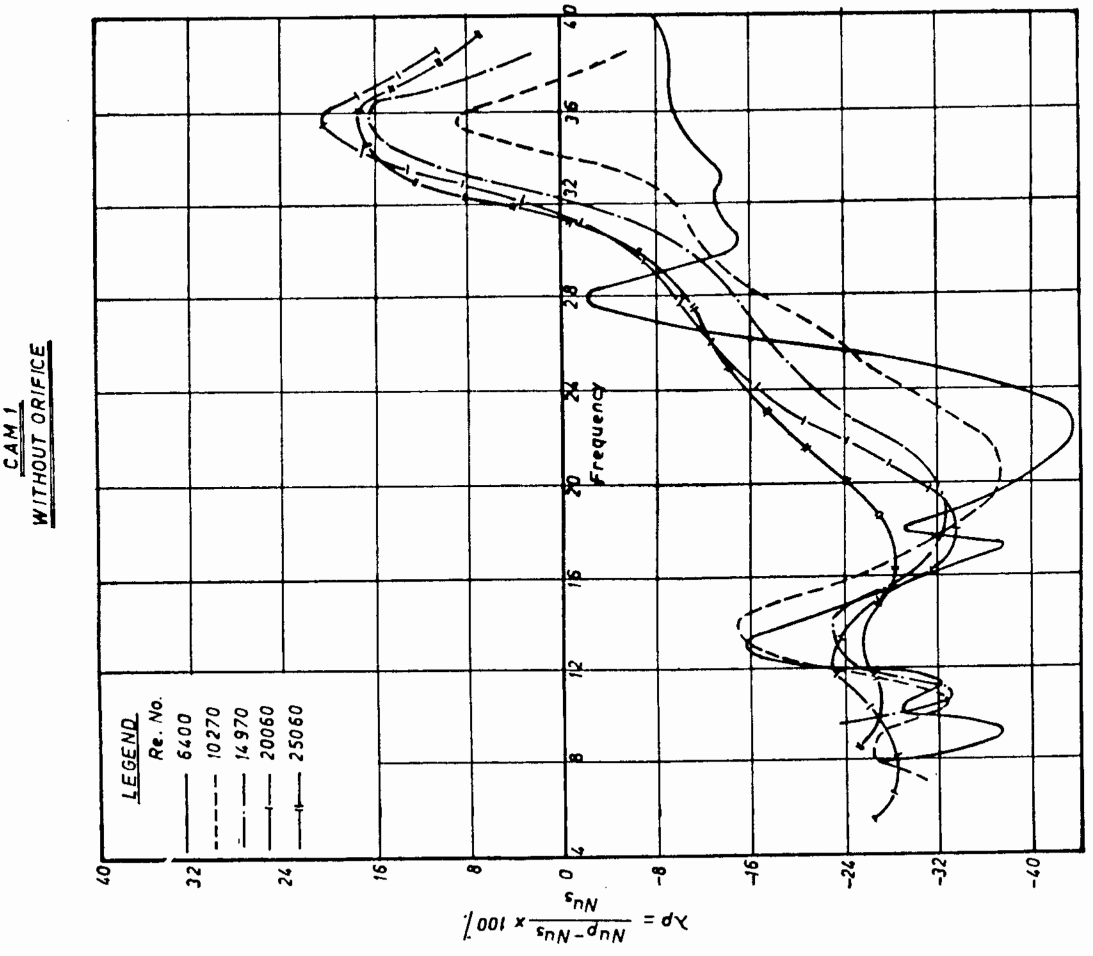

FIG. 13 A

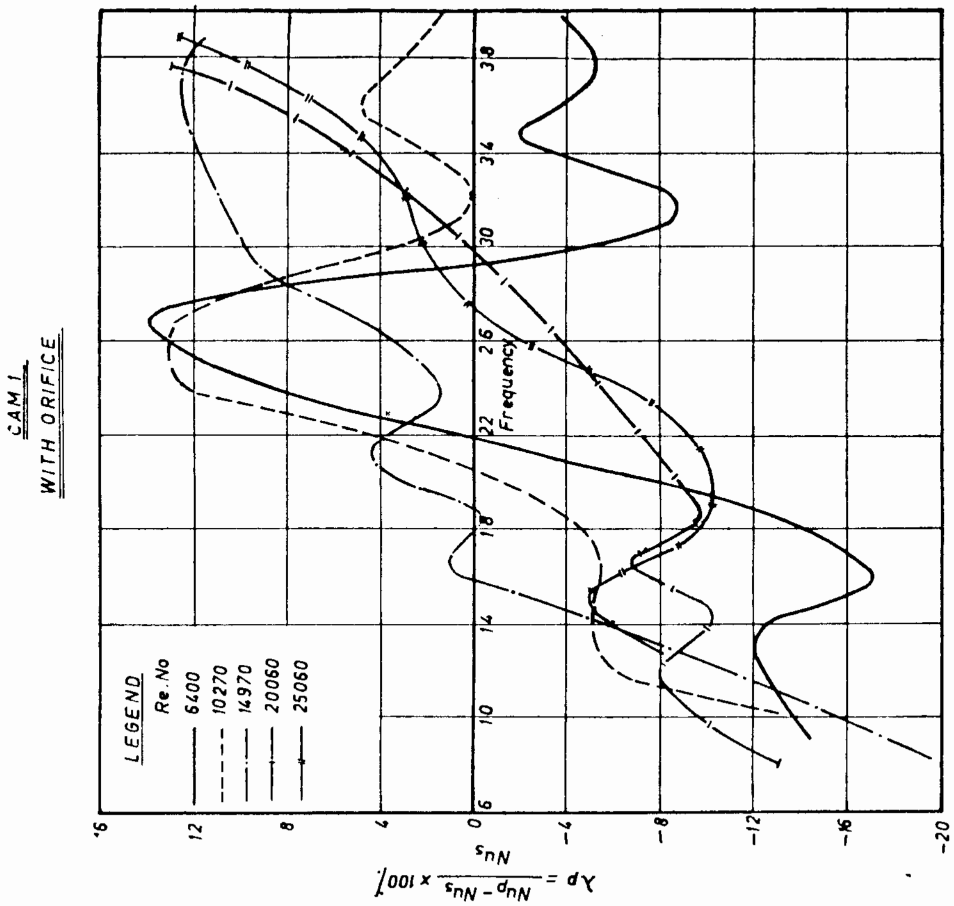

FIG. 13 B

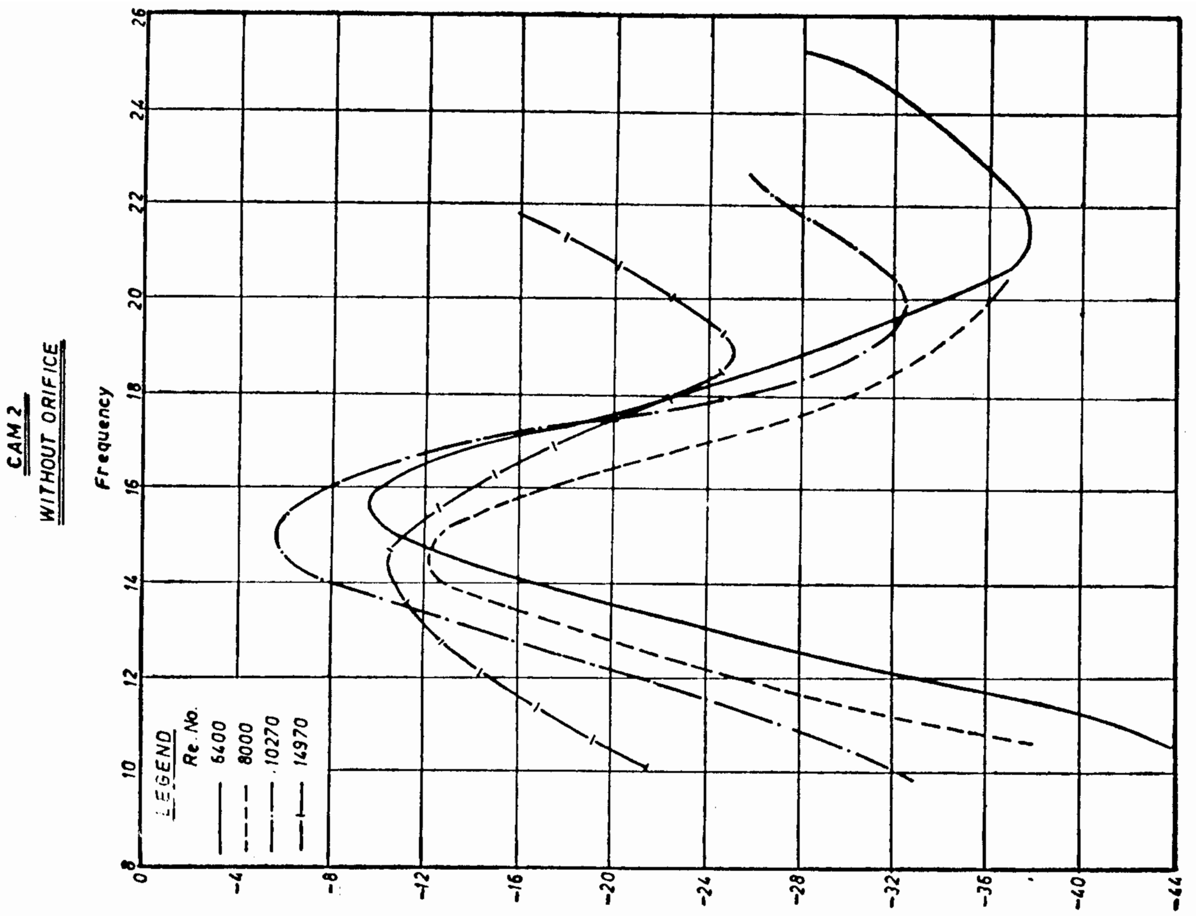

FIG. 13 C

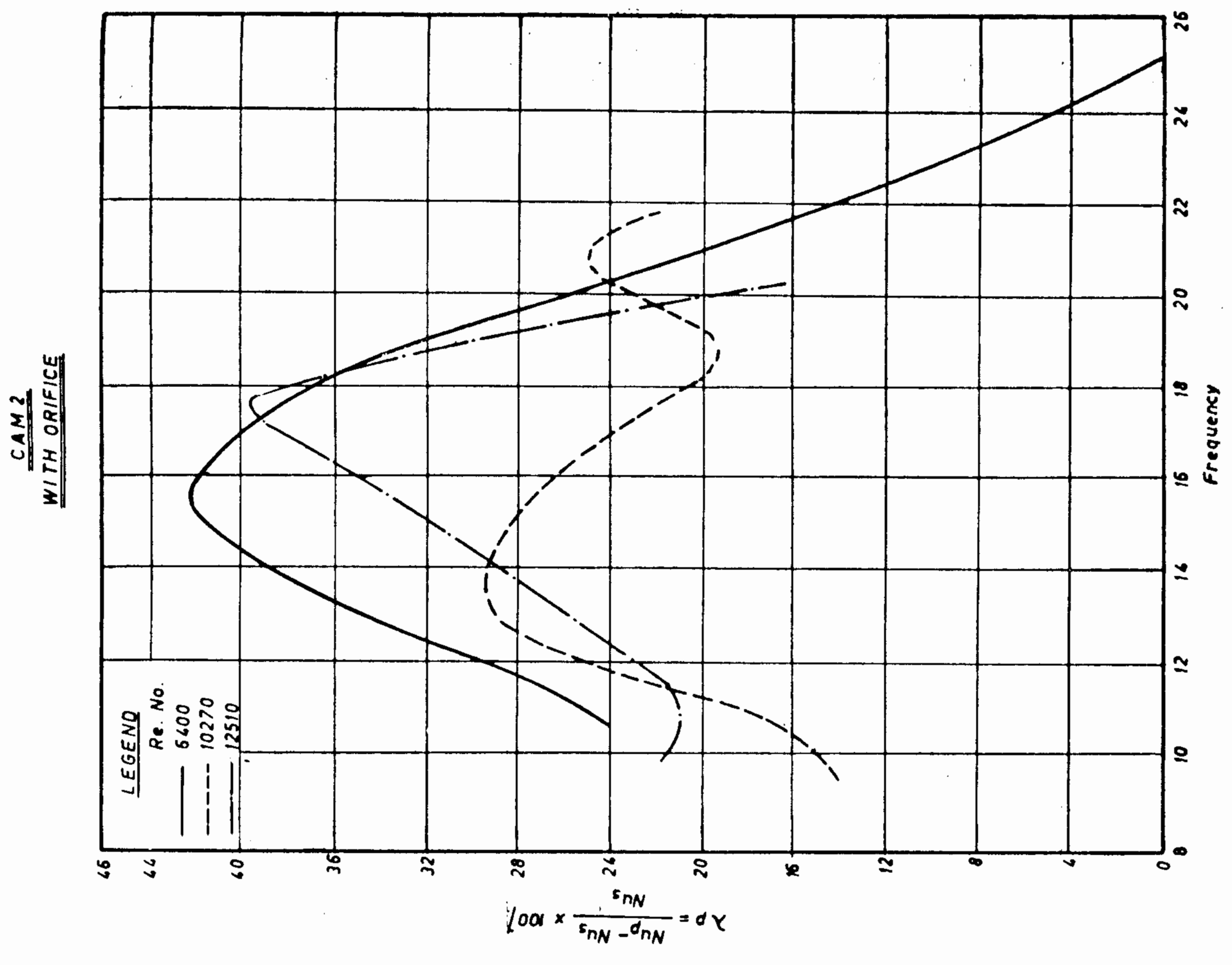

FIG. 13 D

for 80° of cam rotation, no improvement over steady flow heat transfer is obtained (see Fig. 13 C). However, by fitting an orifice plate reducing the free exit area of the heat transfer pipe the first mentioned cam achieves a general improvement above a lower frequency and over a larger range of frequencies (Fig. 13 B). The operation of the second cam results in a general increase of rates of heat transfer for all frequencies investigated (Fig. 13 D).

7. Acknowledgements

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Thanks are due to Mr. N. G. S. Prasanna, Statistical Assistant, for the laborious numerical work required to evaluate the results.

8. List of Symbols and Units, Figures and Tables

8.1. *List of Symbols and Units.*

Symbol	Significance	Units
A	Area of heat transfer tube	m^2
A_1	Area of orifice	m^2
C_d	Coefficient of discharge	..
C_p	Mean specific heat of air at constant pressure	cal./gm. °C.
c/s	Cycles per second	..
D	Inner diameter of heat transfer pipe	m
D_1	Diameter of orifice	m
D_o	Outer diameter of heat transfer section	m
E	$1/\sqrt{(1-m^2)}$, where $m = D_1^2/D^2$	..
g	Gravitational constant	m/s^2
h	Differential head across orifice plate	m of water
H	Rate of heat transfer	cal/s
k	Thermal conductivity	cal/s (m^2) (° C. per m)
L	Effective heating length of heat transfer section	m
n	Frequency	c/s
Nu	Nusselt number	..
Nu_p	Nusselt number for pulsating flow	..
Nu_s	Nusselt number for steady flow	..
Re	Reynolds number	..
T_i	Temperature at inlet to heat transfer section	° C.
T_o	Temperature at outlet of heat transfer section	° C.
T_s	Temperature of formation of steam at atmospheric pressure	° C.
Δt_m	Logarithmic mean temperature	° C.
Δt_{min}	$T_s - T_o$	° C.
Δt_{max}	$T_s - T_i$	° C.
U	Overall coefficient of heat transfer related to outer area of pipe within the heat transfer section	cal./s.m².°C.
V	Velocity of air in heat transfer section	m/s
Q	Rate of flow of air	m^3/s
W	Mass rate of flow of air	g/s
ρ	Specific gravity of air relative to water at the temperature and pressure up-stream of orifice	..
ν	Kinematic viscosity of air	m^2/s
λ_p	$\frac{Nu_p - Nu_s}{Nu_s} \times 100$	%

8.2. *List of Figures.*

8.3. *List of Tables.*

9. List of References

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10. Appendix

Experimental Evaluation

The volume of air flowing through the pipe was calculated from the formula:

$$Q = C_d . A_1 . E . \sqrt{2g} \sqrt{h/\rho}.$$

The velocity in the pipe was then obtained from the formula:

$$V = Q/\pi (D^2/4) \text{ .m/s.}$$

From this the Reynolds number was calculated:

$$Re = VD/\nu.$$

The Reynolds numbers for pulsating flow and for steady flow were calculated in a similar manner in accordance with Martinelli *et al.*[3] Actual numerical calculations showed that for the amount of variation of temperature during experiments with a given mass flow the variation of velocity due to temperature changes was in general negligible, the error in the density being less than 2%.

For calculating the heat transfer coefficient the logarithmic mean temperature differences were used:

$$\Delta t_m = (\Delta t_{max} - \Delta t_{min}) / \log. \Delta t_{max} / \Delta t_{min}.$$

The amount of heat transferred to the air in the heat transfer section is given by

$$H = U . \pi D_0 . L . \Delta t_m,$$

where U = overall coefficient of heat transfer related to outer area of pipe within the heat transfer section.

The amount of heat absorbed by the air is

$$H = C_p W(T_0 - T_i).$$

Equating the two, the heat transfer coefficient is obtained as:

$$U = C_p W(T_0 - T_i)/\pi D_0 L(\Delta t_m).$$

From this the Nusselt number $Nu = UD_0/K$ is obtained.

Nusselt numbers were calculated for steady and pulsating flows at the several Reynolds numbers, giving Nu_s and Nu_p respectively. The percentage difference $(Nu_p - Nu_s)/Nu_s . 100\%$ were then obtained.

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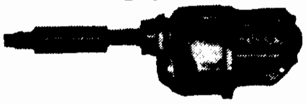

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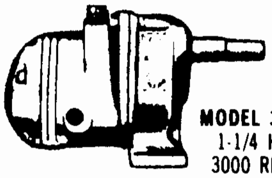

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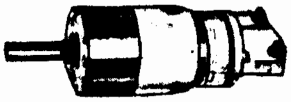

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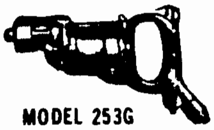

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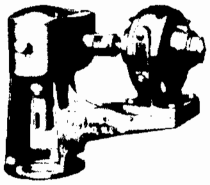

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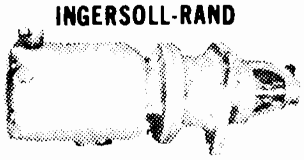

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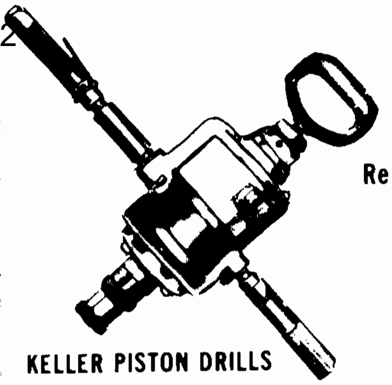

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0	5/16	1800	3/8	5	115
OS1	3/8	900	1/2	6	115
OS2	7/16	625	1/2	6	115
OS3	5/8	375	5/8	6	120
1VH	1/2	1500	1/2	11	155
1V	1/2	1200	1/2	11	175
1VS1	9/16	750	5/8	12	195
1VS2	5/8	500	5/8	12	195
1VS3	7/8	350	5/8	12	200
2VH	29/32	1200	2M.T.	26	260
2V	29/32	700	2	26	260
2VS1	29/32	500	2	26	275
2VS2	1-1/4	400	3	28	275
2VS3	1-1/4	300	3	28	275
3VH	1	580	3	36	280
3V	1	490	3	37	280
3VS1	1-1/4	410	3	37	280
3VS2	1-1/2	330	4	38	305
3VS3	1-1/2	270	4	38	305
3VS4	2	150	4	40	335
3VS5	2-1/4	100	5	40	385

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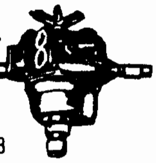

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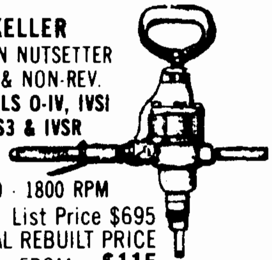

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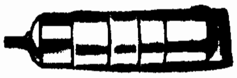
REV. & NON-REV.
MODELS O-IV, IVSI IVS3 & IVSR
350 - 1800 RPM
List Price $695
SPECIAL REBUILT PRICE
FROM **$115**

KELLER SCREWDRIVERS

PISTOL GRIP
NON-REVERSIBLE
Takes Standard Hex Bits
SPECIAL REBUILT PRICE **$24.95**

ARO



MODEL 7819
650 RPM
List Price $605
SPECIAL REBUILT PRICE **$75**

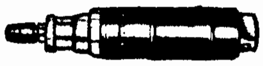
INGERSOLL-RAND

MODEL E9G
5.4 HP - 130 RPM
NON-REVERSIBLE
List Price $1995
SPECIAL REBUILT PRICE **$535**

CLECO

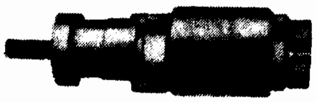
MODEL 401-NS
460 RPM
List Price $560
SPECIAL REBUILT PRICE **$115**

THOR



MODEL 5M
800 RPM
List Price $360
SPECIAL REBUILT PRICE **$195**

INGERSOLL-RAND



MODEL 2P
230 RPM
List Price $565
SPECIAL REBUILT PRICE **$155**

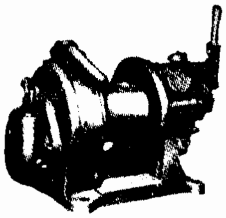
ARO

MODEL 7256 BS
1/3 H.P. - 750 R.P.M.
3/8" - 24 Thd. Shaft
New $230 - REBUILT **$115**

REBUILT

AIR TUGGERS

GUARANTEED

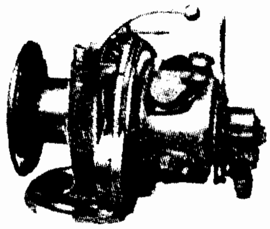


INGERSOLL — RAND TUGGER

LIST $2905

Size BU 750 lb Capacity Lifting Speed 40 Ft Per Min Weight 75 lbs. Cable Capacity 350 Ft of 3/16"

Special Rebuilt Price **$1350.00**

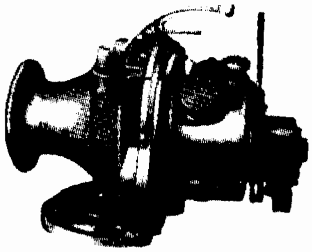


INGERSOLL — RAND TUGGER

LIST $4690

Size DUU 1000 lb Capacity Lifting Speed 75 Ft Per Min Weight 275 lbs Cable Capacity 290 Ft of 5/16"

Special Rebuilt Price **$1975.00**



INGERSOLL — RAND TUGGER

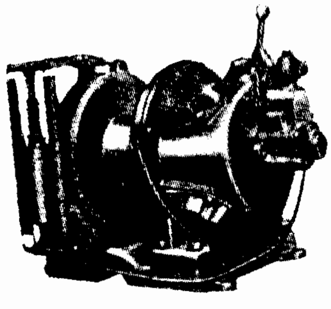
LIST $4145

Size DU 750 lb Capacity Lifting Speed 75 Ft Per Min Weight 270 lbs Cable Capacity 290 Ft of 5/16"

Rebuilt Price **$1475.00**

INGERSOLL — RAND TUGGER

LIST $6285

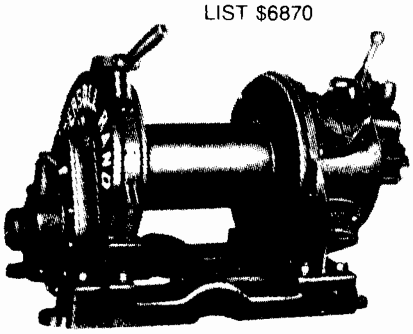


Size HU, 2000 lb Capacity
Lifting Speed 124 Ft Per Min
Weight 490 lbs
Cable Capacity 400 Ft of 7/16"

Special Rebuilt Price **$2995.00**

INGERSOLL — RAND TUGGER

LIST $6870

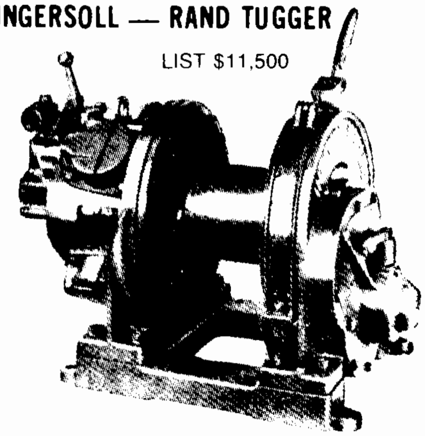


Size HUL, 2000 lb Capacity
Lifting Speed 124 Ft Per Min
Weight 562 lbs
Cable Capacity 850 Ft of 7/16"

Special Rebuilt Price **$3295.00**

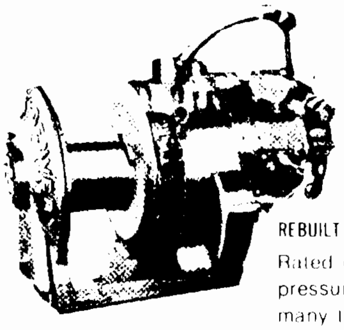
INGERSOLL — RAND TUGGER

LIST $11,500



Size K4U, 4000 lb. Capacity
Lifting Speed 95 Ft Per Min
Cable Capacity 550 Ft of 1/2"
Weight 850 lbs

Special Rebuilt Price **$5995.00**



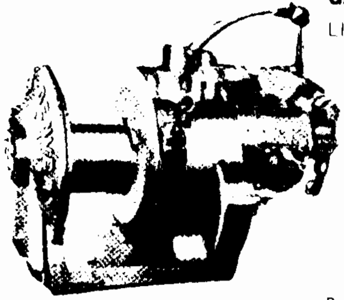
GARDNER DENVER

LIST $4600

Model HE Tugger and Utility Hoist for continuous heavy duty service

REBUILT **$2295.00**

Rated capacity 2500 lbs at 80 lbs air pressure Capacity can be increased many times by use of sheave or block and tackle



GARDNER DENVER

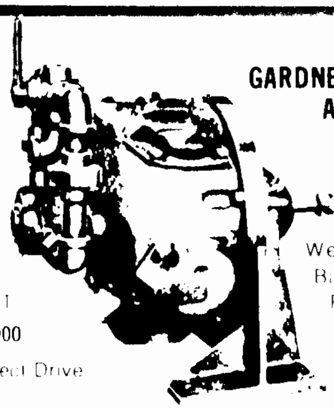
LIST $2425

Model HB

Rated capacity 1000 lbs at 80 lbs air pressure

Capacity can be increased many times by use of sheave or block tackle

Barrys Special Rebuilt Price **$1150.00**



GARDNER-DENVER AIR MOTOR

Reversible
Model MKB
2500 RPM
10 H P
Weight 180 lbs
Barrys Special Rebuilt Price
$715.00

LIST
$1900
Direct Drive

INGERSOLL-RAND AIR MOTOR

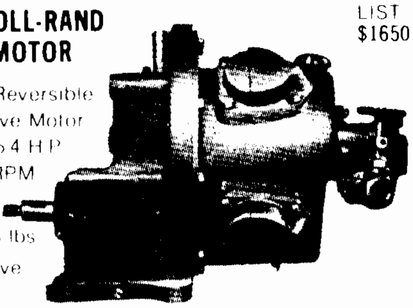
Size CM 2.9 H P
LIST
$1250

Direct Drive

1400 to 3200 RPM
Weight 130 lbs
Special Rebuilt Price **$650.00**

INGERSOLL-RAND AIR MOTOR

LIST
$1650



E9G Non-Reversible Geared Drive Motor
Size E9G 5.4 H P
130 - 285 RPM

Weight 205 lbs
Geared Drive

Barrys Special Rebuilt Price **$835.00**

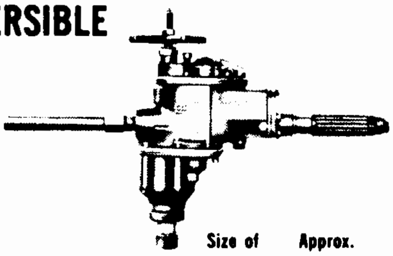
HEAVY DUTY DRILLS

REBUILT

1/2"TO 3"CAPACITY

Heavy Duty Air Drills are preferred for production work because they have the absolute smoothness of operation that is so necessary for accurate drilling, reaming, tapping, nut running, etc. At the same time, they possess the power to keep them up to speed when under a load. They are light in weight and easy to handle. Their power and efficiency make them ideal tools for many jobs that formerly required larger and heavier machines.

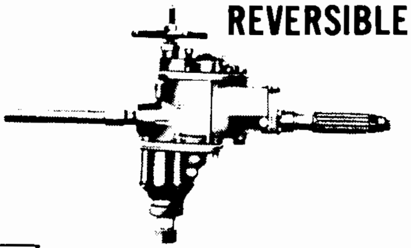
NON-REVERSIBLE



CAP.	Manufacturer	Model	RPM	Size of M.T.	Approx. List	Rebuilt
1/2"	CHICAGO PNEUMATIC	310-P	925	1 or Chuck	$2200.00	$ 495.00
	CHICAGO PNEUMATIC	310-S	925	1 or Chuck	2200.00	495.00
9/16"	CHICAGO PNEUMATIC	315	750	1 or Chuck	2460.00	650.00
	CHICAGO PNEUMATIC	323-S	550	1 or Chuck	2460.00	650.00
	CHICAGO PNEUMATIC	323-S	700	1 or Chuck	2460.00	650.00
	CHICAGO PNEUMATIC	315	500	1 or Chuck	2460.00	650.00
	INGERSOLL RAND	2XL	500	2	2370.00	650.00
	INGERSOLL RAND (New)	2XK	725	1	2370.00	775.00
	THOR	253G	700	Chuck	2370.00	625.00
	THOR	253X	700	1	2370.00	625.00
	THOR	353X	700	1	2370.00	675.00
	THOR	353Y	550	1	2370.00	675.00
7/8"	CHICAGO PNEUMATIC	323-S	350	1 or Chuck	2875.00	700.00
	CHICAGO PNEUMATIC	315	315	1 or Chuck	2875.00	725.00
	INGERSOLL RAND	2XM	350	2	2370.00	725.00
	THOR	353Z	350	1	2370.00	675.00
29/32"	CHICAGO PNEUMATIC	327	700	2	3750.00	950.00
	CHICAGO PNEUMATIC	326	700	2 or 3	3750.00	900.00
	INGERSOLL RAND	3H	800	2	3890.00	1075.00
	INGERSOLL RAND	3J	450	2	3890.00	1075.00
	THOR	362H2	700	2	3790.00	900.00
1"	CHICAGO PNEUMATIC	326	450	2	4160.00	1025.00
	CHICAGO PNEUMATIC	327	450	3	4160.00	1075.00
	INGERSOLL RAND	3SJ	450	3	4260.00	1075.00
	INGERSOLL RAND	3SH	800	3	4260.00	1075.00
	THOR	362X	450	3	4260.00	1025.00
1-1/4"	CHICAGO PNEUMATIC	327	375	3	4890.00	1175.00
	INGERSOLL RAND	4J	450	3	4900.00	1175.00
	INGERSOLL RAND	M4J	450	4	4900.00	1175.00
	INGERSOLL RAND	3SM	185	3	4200.00	1650.00
	INGERSOLL RAND	4K	310	3	4900.00	1275.00
	INGERSOLL RAND	3SK	300	3	3900.00	1175.00
	ROTOR	884	200	4	3490.00	925.00
	THOR	363Z	250	3	3690.00	1050.00
	THOR	363Y	350	3	3690.00	1075.00
	THOR	363X3	450	3	3690.00	1100.00
2"	CHICAGO PNEUMATIC	350	275	4	5175.00	1775.00
	CHICAGO PNEUMATIC	350	450	4	5175.00	1775.00
	INGERSOLL RAND	4SM	160	4	5150.00	1800.00
	INGERSOLL RAND	M5J	450	4	5150.00	1750.00
	THOR	385Y	300	4	4890.00	1375.00
	THOR	385Y3	300	3	4890.00	1375.00
2-1/2"	CHICAGO PNEUMATIC	350	150	4	5175.00	1800.00
	INGERSOLL RAND	5-SM	120	5	5175.00	1900.00

SPECIAL DRILLS AVAILABLE
OTHER MODELS AVAILABLE

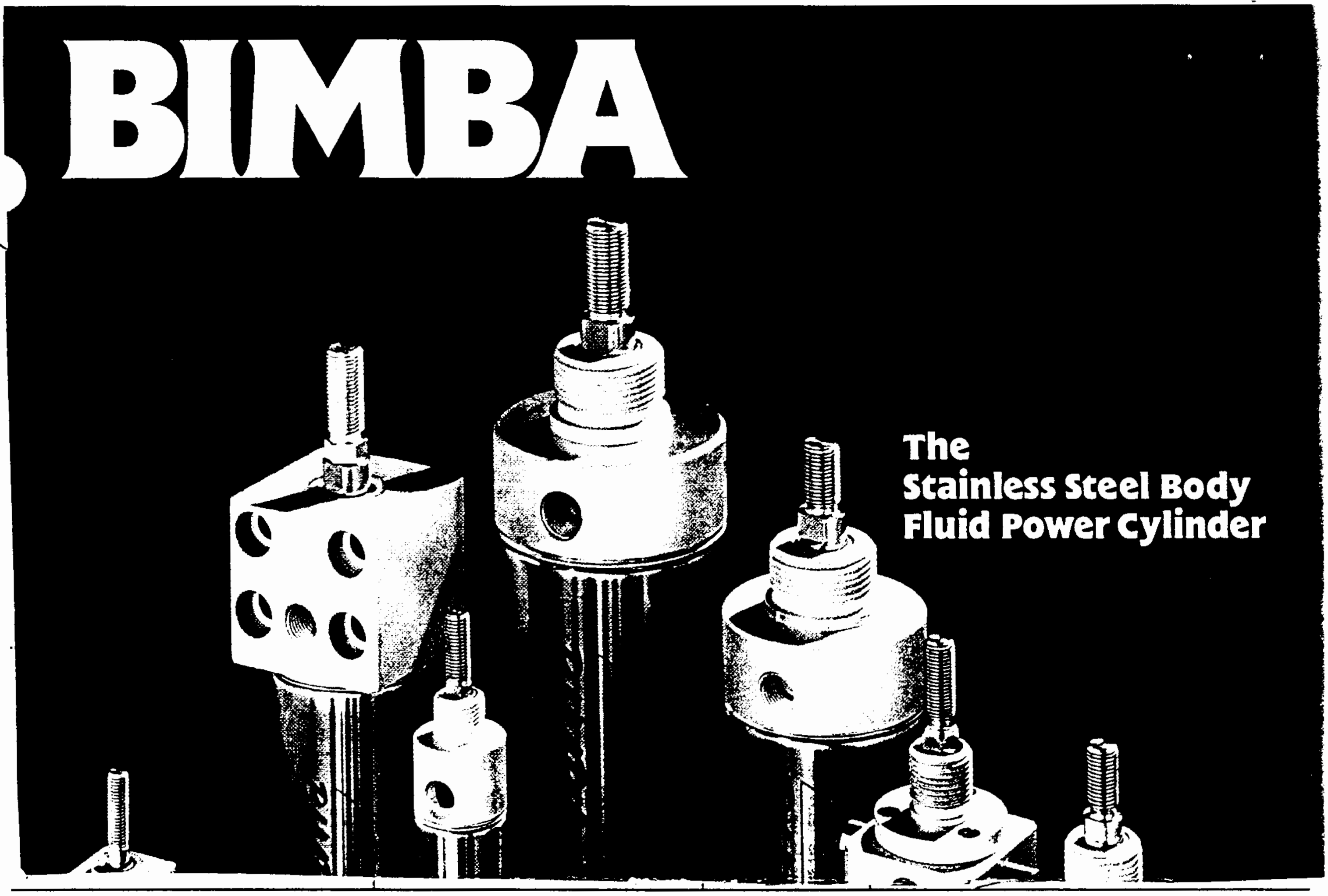
REVERSIBLE



CAP	Manufacturer	Model	RPM	Size of M.T.	Approx. List	Rebuilt
9/16"	CHICAGO PNEUMATIC	315R	750	2	$2350.00	$ 700.00
	INGERSOLL RAND	22H	1500	Chuck	2390.00	725.00
	INGERSOLL RAND	22K	725	1 or 2	2390.00	725.00
	ROTOR	40C	700	2	1965.00	585.00
	ROTOR	20	500	1	1965.00	585.00
	THOR	353RX	700	1 or 2	1965.00	675.00
7/8"	CHICAGO PNEUMATIC	315R	200	3	2415.00	875.00
	CHICAGO PNEUMATIC	315R	350	2	2415.00	875.00
	CLECO	101AR	300	2	1890.00	695.00
	INGERSOLL RAND	22L	550	2	2390.00	750.00
	THOR	353RZ	200	3	2360.00	725.00
15/16"	CHICAGO PNEUMATIC	315R	100	3	2515.00	895.00
29/32"	CHICAGO PNEUMATIC	327R	700	2 or 3	4160.00	1025.00
	CHICAGO PNEUMATIC	326R	650	3	4160.00	950.00
	INGERSOLL RAND	33J	450	2	4160.00	1250.00
	THOR	362RH2	700	2	4160.00	950.00
1"	CHICAGO PNEUMATIC	327R	500	3	4290.00	1150.00
	INGERSOLL RAND	33SH	800	3	4290.00	1150.00
	INGERSOLL RAND	33SJ	450	3	4290.00	1150.00
	THOR	362RX3	450	3	4190.00	1075.00
	THOR	263RM	450	3	4190.00	1050.00
	THOR	363RY3	350	3	4190.00	1175.00
1-1/4"	CHICAGO PNEUMATIC	327R	400	3	4390.00	1300.00
	CHICAGO PNEUMATIC	327R	290	3	4390.00	1300.00
	CHICAGO PNEUMATIC	327R	160	4	4555.00	1825.00
	CLECO	103AR	160	3	4555.00	1250.00
	INGERSOLL RAND	44SL	225	4	5300.00	1850.00
	INGERSOLL RAND	33SM	185	3 or 4	4345.00	1850.00
	INGERSOLL RAND	33M4	185	4	4210.00	1850.00
	INGERSOLL RAND	44J	450	3	5300.00	1275.00
	INGERSOLL RAND	33SK	300	3	4290.00	1275.00
2"	CHICAGO PNEUMATIC	350R	275	4	5475.00	1880.00
	CHICAGO PNEUMATIC	350R	450	4	5475.00	1880.00
	CHICAGO PNEUMATIC	350R	100	4	5475.00	1900.00
	INGERSOLL RAND	M55K	300	4	5775.00	1975.00
	INGERSOLL RAND	44SM	160	4	5450.00	1925.00
	THOR	385RY	310	4	5190.00	1425.00
	THOR	264RY	130	4	5190.00	1375.00
2-1/2"	CHICAGO PNEUMATIC	327R	55	4	5815.00	1975.00
	INGERSOLL RAND	55L	200	4	5775.00	2075.00
3"	CHICAGO PNEUMATIC	350R	100	5	5815.00	1675.00
	INGERSOLL RAND	55-SM	120	5	5775.00	1975.00
	THOR	385RL3	125	3	5410.00	1600.00
OVER 3"	CHICAGO PNEUMATIC	350R	65	5	5815.00	1900.00
	INGERSOLL RAND	55R	25	5	5775.00	2075.00
	THOR	385RL5	75	5	5775.00	1875.00

ALL PRICES QUOTED ARE F.O.B. OUR DETROIT PLANT

BARRY AIR TOOL SALES • 10650 CLOVERDALE • DETROIT, MICHIGAN 48204

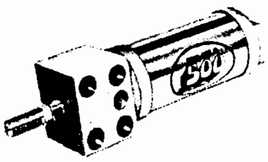
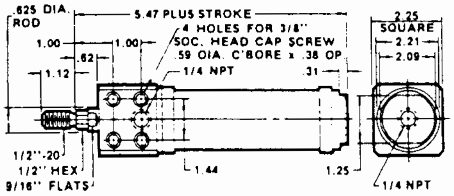
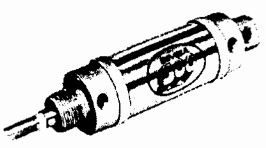
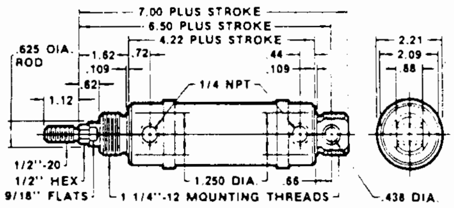
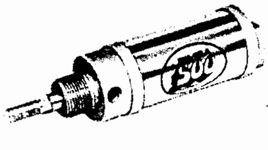
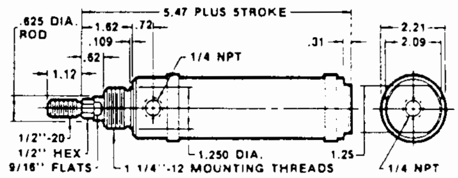


2" BORE

☐ Enter Stroke Length as 3rd Digit

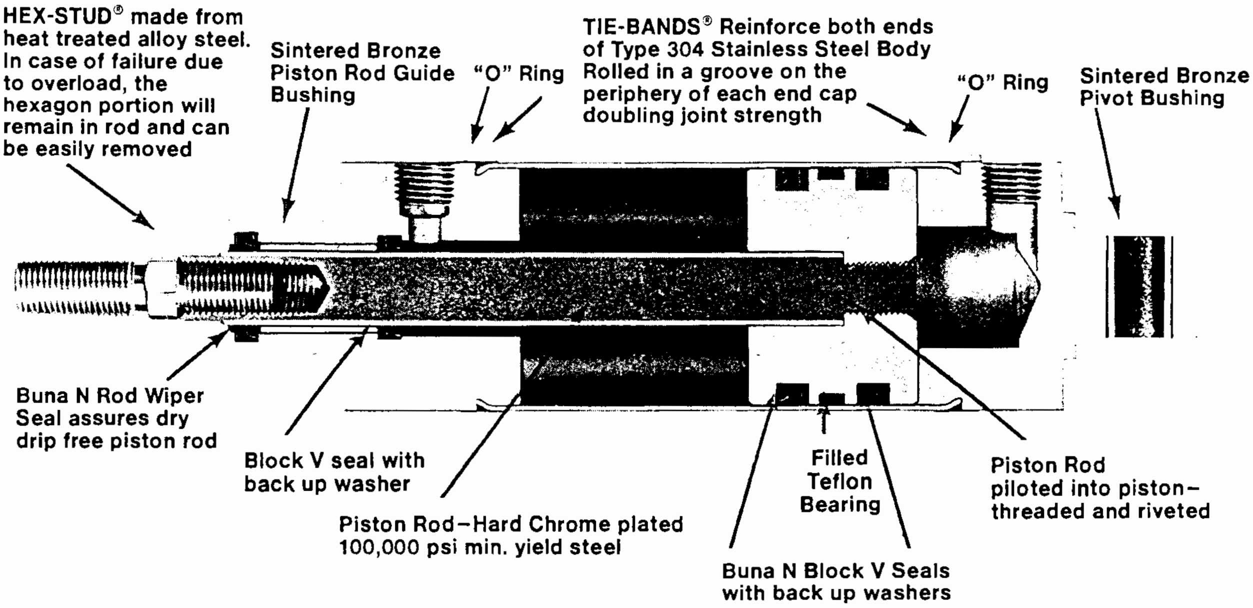
"500" HYDRAULIC CYLINDERS

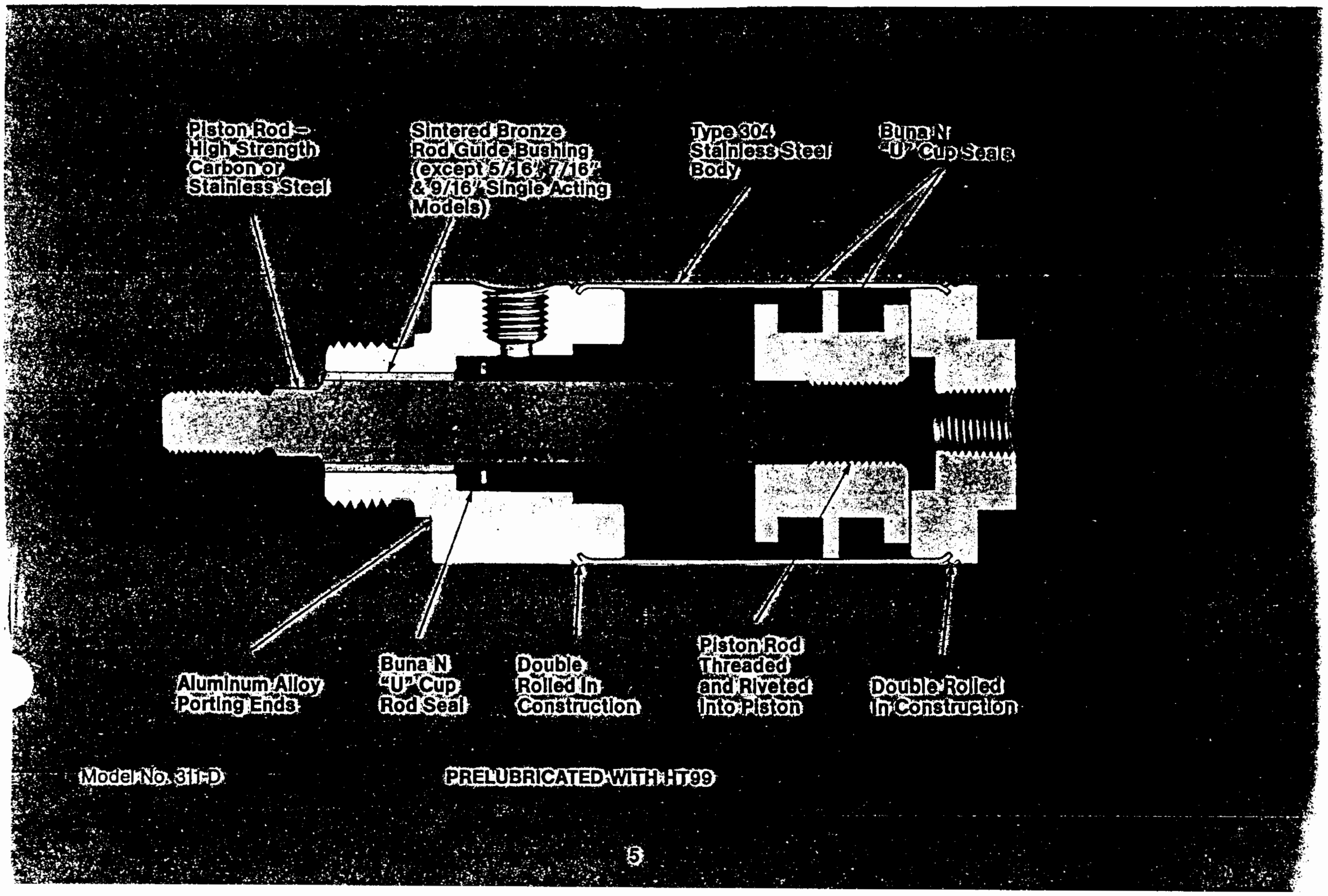
- Push Force – 3.14 x psi, Pull Force – 2.83 x psi
- Rated 500 psi Hydraulic (non-shock), Double Acting

MODEL	DESCRIPTION	PRICE	DIMENSIONS
H-31☐-DBZ 	**Front Block Mount** Standard Stroke Lengths: 1" increments to 6" Maximum Stroke 12"	**$84.25** BASE PRICE Add $2.30 per inch of stroke	
H-31☐-DUZ 	**Universal Mount**–Double End or Rear Pivot– Standard Stroke Lengths: 1" increments to 12" Maximum Stroke 32" Optional Accessories: D-8325-A Pivot Bracket D-8319 Mounting Bracket D-8313-A Clevis	**$78.90** BASE PRICE Add $2.30 per inch of stroke	
H-31☐-DZ 	**Nose Mount** Standard Stroke Lengths: 1" increments to 6" Maximum Stroke 12" Optional Accessory: D-8319 Mounting Bracket	**$73.65** BASE PRICE Add $2.30 per inch of stroke	

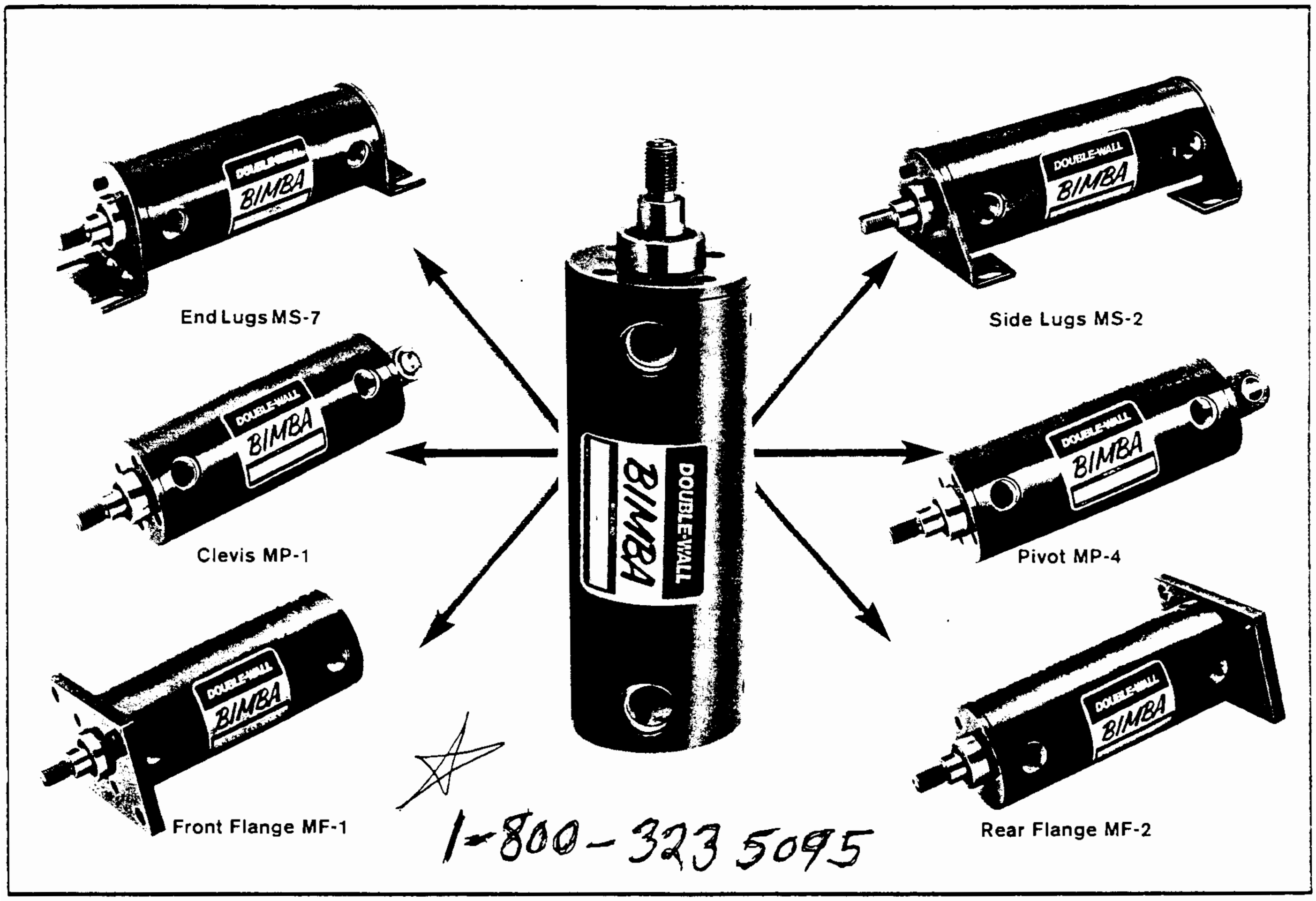
500®

HYDRAULIC CYLINDERS

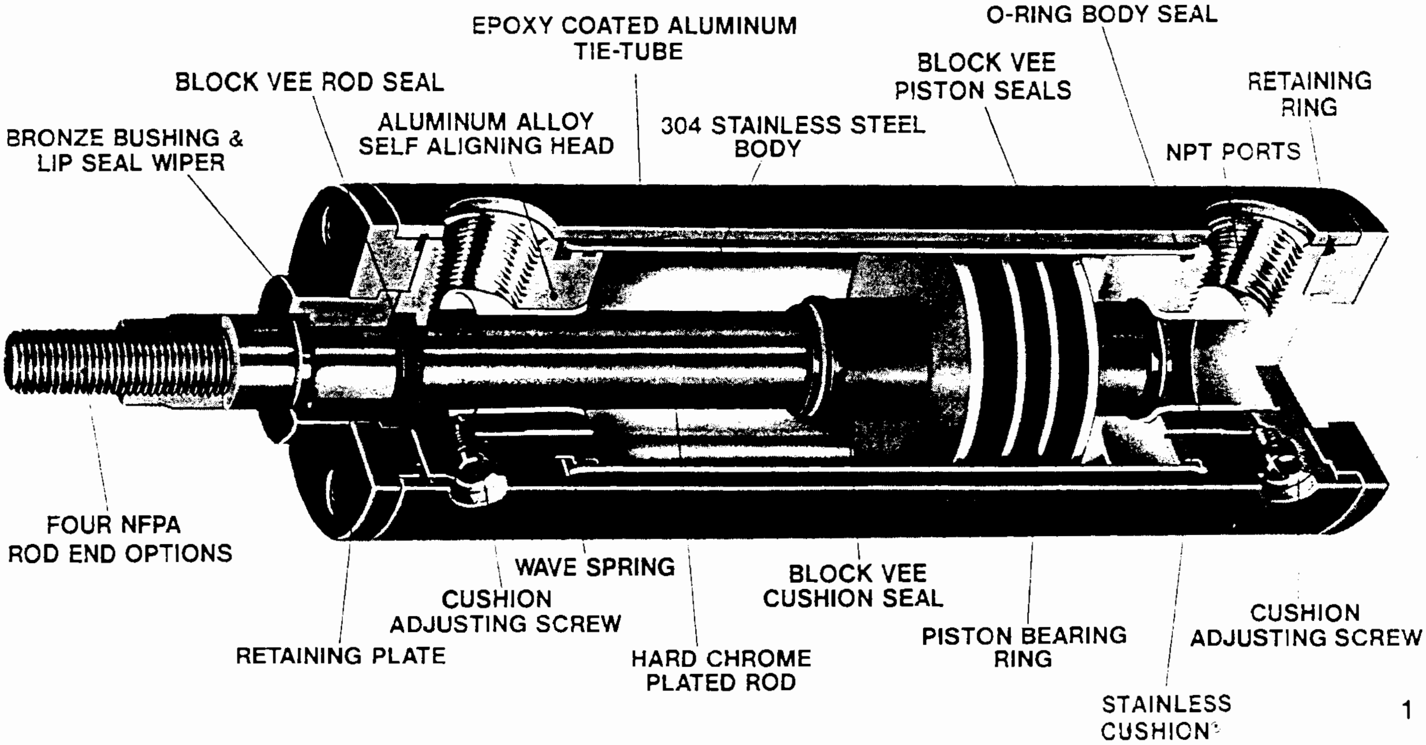




REPLACE THOSE BULKY TIE-ROD CYLINDERS WITH THE STREAMLINED BIMBA DOUBLE-WALL® TIE-TUBE DESIGN AIR CYLINDER. ONE BASIC CYLINDER CONVERTS INTO SIX NFPA MOUNTING STYLES.



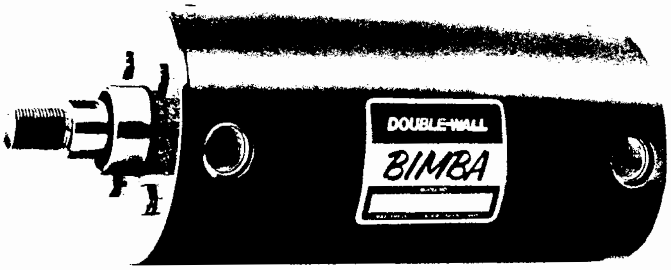
ANATOMY OF THE BIMBA DOUBLE-WALL®



4" BORE DOUBLE-WALL® SERIES DW-125

- TIE-TUBE - Epoxy Coated Aluminum
- BODY - Smooth I.D. Type 304 Stainless Steel
- PRESSURE RATING - 200 PSI Air
- POWER FACTOR - 12.5 of Air Line Pressure
- PISTON ROD - Hard Chrome Plated, Standard
- BUNA N SEALS - Viton see Page 9 for Prices
- CUSHIONS — Exclusive Bimba STAINLESS CUSHION®. Available Either or Both Ends, see Page 2 "How to Order" and Page 9 for Prices
- STROKE LENGTHS - 1" Increments Thru 24" Standard, Long and Fractional Strokes Available on Request
- ROD END OPTIONS - Four NFPA Rod Ends Available, see Page 2 for Price and Dimensions. See Important Information, Inside Back Cover for Special Rod and Rod End Availability. One-Piece Threaded Male Rod (style #2) Shown Below is Shipped Unless Otherwise Specified.
- MOUNTING KITS - Six NFPA Mountings Available in Kits (including necessary hardware) for Attachment to Basic Cylinder. MOUNTING KITS AND BASIC CYLINDERS ARE ORDERED AND SHIPPED AS SEPARATE ITEMS.

BASIC DOUBLE-WALL® CYLINDER

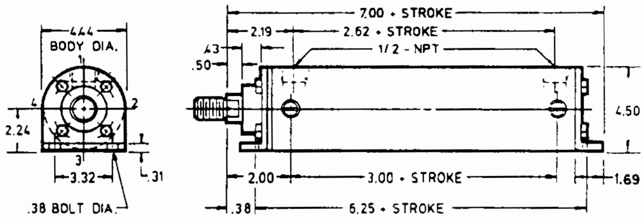


BASE PRICE **$134.00** PLUS $4.55 per inch of stroke

(See Page 2 – How To Order)

MODEL	DESCRIPTION	PAGE
4" BORE MOUNTING KITS		
MSL-125	Side Lug Mount	7
MEL-125	End Lug Mount	7

END LUG MOUNTS (NFPA MS-7)

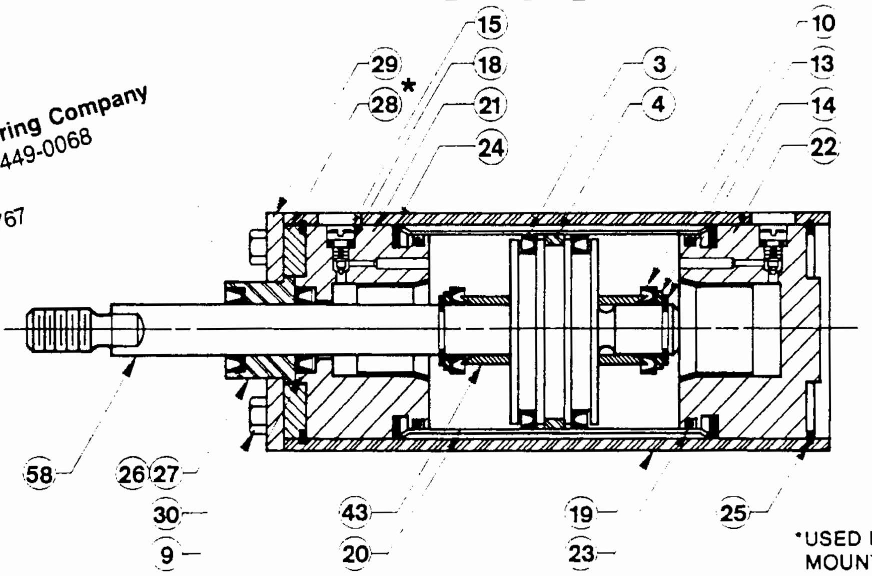


MOUNTING KIT **#MEL-125** PRICE **$20.35**

BASIC DOUBLE-WALL® CYLINDER DRAWINGS

DW SERIES

Bimba Manufacturing Company
Monee, Illinois 60449-0068
708/534-8544
FAX: 708/534-5767



Bimba Limited
23 Maxwell Road
Woodston, Peterborough
Cambridgeshire PE2 7JD
United Kingdom
0733 391078
FAX: 0733 391080

*USED IN MOST MOUNTING KITS AND ON ALL 3¼" AND 4" BORE BASIC CYLINDERS.

from a fluid power manual by Anthony Esposito

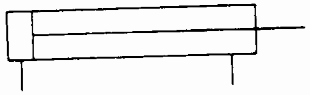
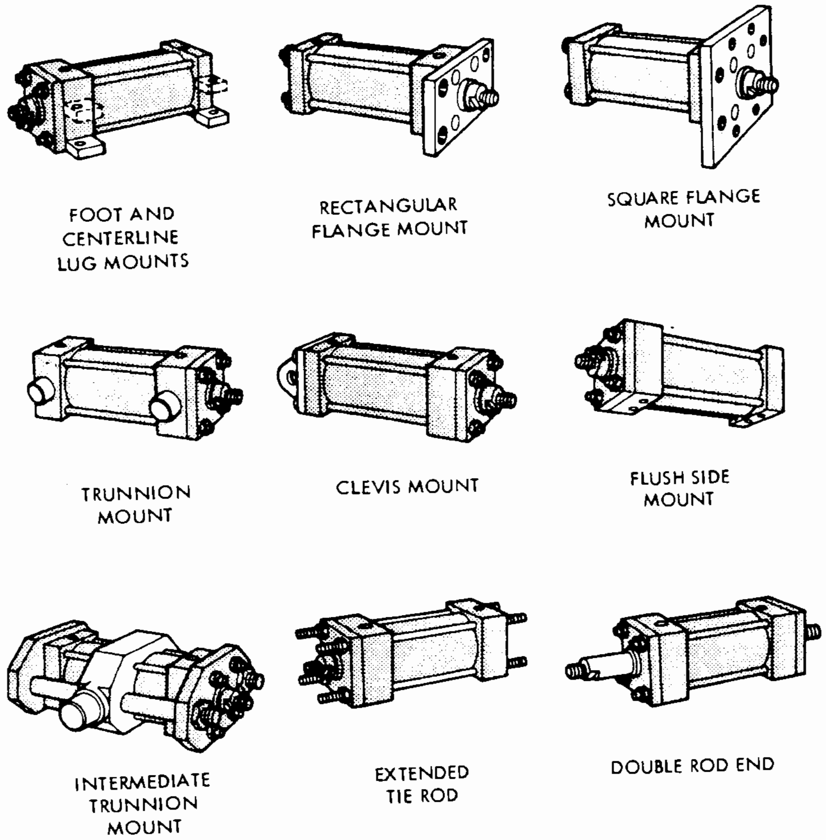


Fig. 7-3. Symbolic representation of double-acting cylinder. (Courtesy of Sheffer Corporation, Cincinnati, Ohio.)

smooth deceleration at both ends of the stroke. Therefore, the piston does not bang into the end caps with excessive impact, which could damage the hydraulic cylinder after a given number of cycles. The symbol for a double-acting cylinder is shown in Fig. 7-3. Notice that the symbol implies how the cylinder operates without showing any details. In drawing hydraulic circuits (as done in Chapter 9), symbolic representation of all components will be used. This facilitates circuit analysis and troubleshooting. Also, it would be too time-consuming to draw each component schematically. The symbols, which are merely combinations of simple geometric figures such as circles, rectangles, and lines, make no attempt to show the internal configuration of a component. However, symbols must clearly show the function of each component.

Various types of cylinder mountings are in existence, as illustrated in Fig. 7-4. This permits versatility in the anchoring of cylinders. The rod ends are usually

Fig. 7-4. Various cylinder mountings. (Courtesy of Sperry Vickers, a Division of Sperry Rand Corp., Troy, Michigan.)



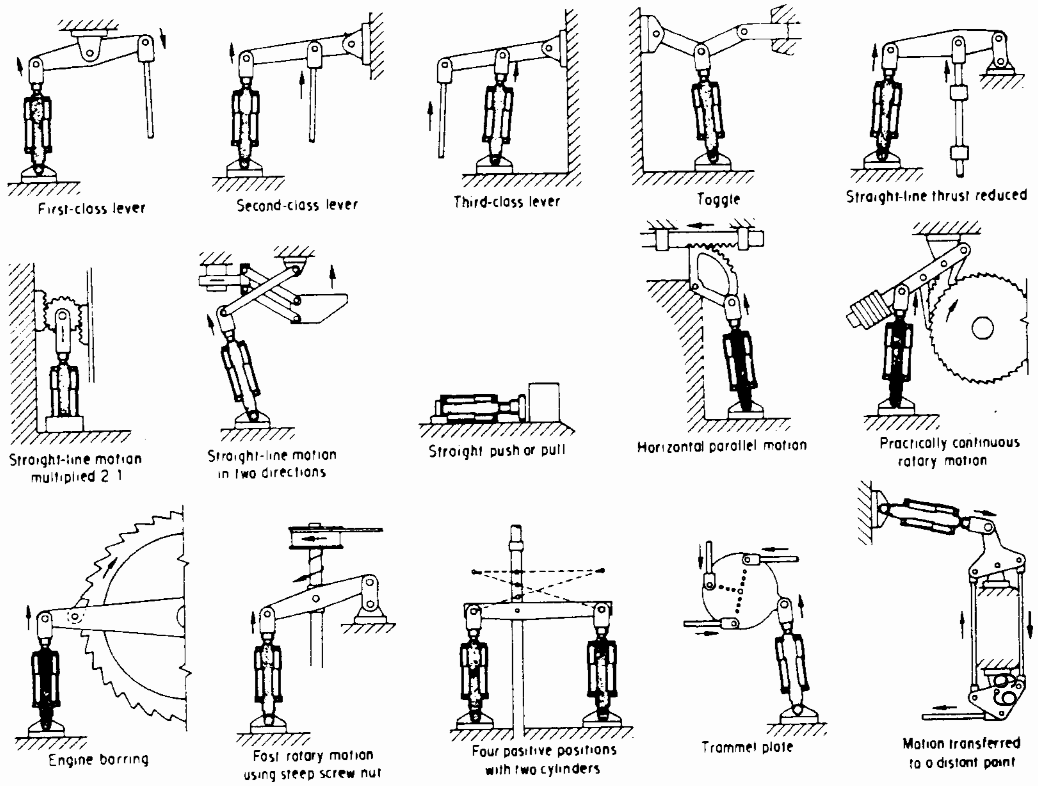


Fig. 7-5. Typical mechanical linkages that can be combined with hydraulic cylinders. (Courtesy of Rexnord Inc., Hydraulic Components Division, Racine, Wisconsin.)

threaded so that they can be attached directly to the load, a clevis, a yoke, or some other mating device.

Through the use of various mechanical linkages, the applications of hydraulic cylinders are limited only by the ingenuity of the fluid power designer. As illustrated in Fig. 7-5, these linkages can transform a linear motion into either an oscillating or rotary motion. In addition, linkages can also be employed to increase or decrease the effective leverage and stroke of a cylinder.

Much effort has been made by manufacturers of hydraulic cylinders to reduce or eliminate the side loading of cylinders created as a result of misalignment. It is almost impossible to achieve perfect alignment even though the alignment of a hydraulic cylinder has a direct bearing on its life.

A universal alignment mounting accessory designed to reduce misalignment problems is illustrated in Fig. 7-6. By using one of these accessory components and a

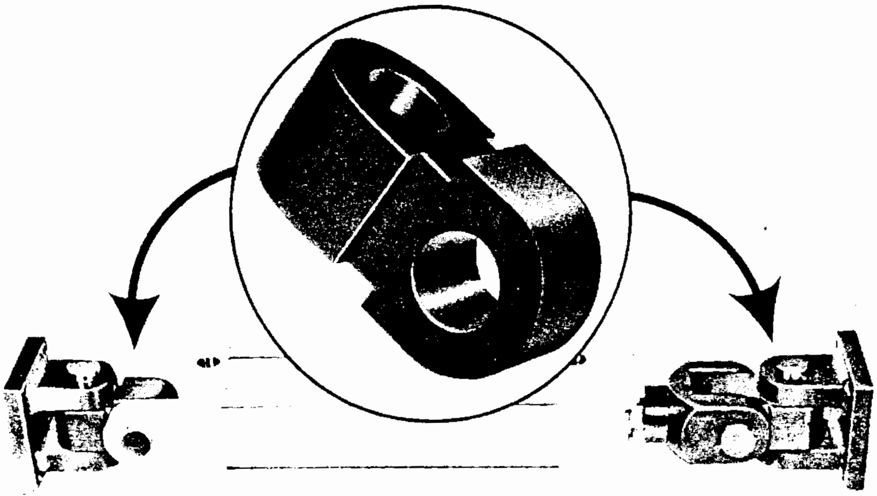


Fig. 7-6. Universal alignment mounting accessory for fluid cylinders. (Courtesy of Sheffer Corporation, Cincinnati, Ohio.)

mating clevis at each end of the cylinder (see Fig. 7-6), the following benefits are obtained:

1. Freer range of mounting positions
2. Reduced cylinder binding and side loading
3. Allowance for universal swivel
4. Reduced bearing and tube wear
5. Elimination of piston blow-by caused by misalignment

The force output and piston velocity of double-acting cylinders is not the same for extension and retraction strokes. This phenomenon is due to the effect of the rod and is defined by Eqs. (7-1)–(7-4).

Extension stroke:

$$\text{force (lb)} = \text{pressure(psi)} \times \text{piston area (in.}^2\text{)} \tag{7-1}$$

$$\text{velocity (ft/sec)} = \frac{\text{input flow (ft}^3\text{/sec)}}{\text{piston area (ft}^2\text{)}} \tag{7-2}$$

Retraction stroke:

$$\text{force (lb)} = \text{pressure (psi)} \times [\text{piston area (in.}^2\text{)} - \text{rod area (in.}^2\text{)}] \tag{7-3}$$

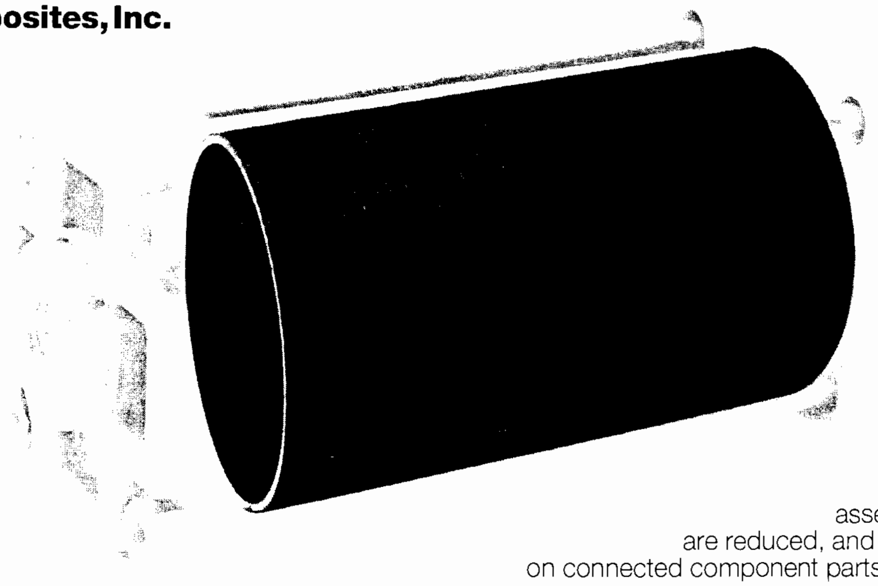
$$\text{velocity (ft/sec)} = \frac{\text{input flow (ft}^3\text{/sec)}}{\text{piston area (ft}^2\text{)} - \text{rod area (ft}^2\text{)}} \tag{7-4}$$

Black-Amalgon™

HYDRAULIC AND PNEUMATIC CLYLINDER TUBING

The Better Choice...

from Amalga Composites, Inc.



Black-Amalgon™, the Alternative to Metal

For more than 20 years, there has been an alternative to metallic pneumatic and hydraulic cylinder tubing for low-, medium- or high-pressure systems: Black-Amalgon™.

Constructed of fiber-reinforced thermoset epoxy matrix, Black-Amalgon has an inner layer of evenly dispersed low-friction additives. The result: A light-weight, high-strength, corrosion-resistant composite material which can easily replace carbon steel, honed and chromed steel, stainless steel, aluminum, or brass cylinder barrels.

Black-Amalgon for low pressure applications is constructed of a fiberglass/epoxy composite. Depending upon the pressure requirements, medium pressure and high pressure versions may incorporate other materials in hybrid configurations.

Four Big Reasons

1. **75% Reduction in Weight** — Right off the bat, Black-Amalgon saves you money in material costs, handling, shipping and storage. At approximately 1/4 the weight of steel or brass, and 3/4 the weight of aluminum, Black-Amalgon is much easier to handle than traditional metal tubing. Therefore, freight costs are lower, assembly times are reduced, and stress loads on connected component parts decrease.
2. **Superior Corrosion Resistance** — You can expect trouble-free performance in chemical, high moisture and other adverse environments that would normally corrode or impair the operation of metals. No problem with salt or chlorinated water either. With Black-Amalgon, you'll see significant reduction in life-cycle costs.
3. **Reduced Maintenance Costs** — Say "good bye" to piston lock-up problems. Black-Amalgon's patented manufacturing process ensures a smooth, self-lubricating, homogeneous internal surface that prevents pistons from sticking, even after they've remained idle for some time.
4. **Eliminate Honing Costs** — A surface smoother than honed steel...without the cost of honing. A 5-15 RMS micro-inch ID finish performs just like a honed or chromed surface.

Chemical Resistance

Available Cylinder Tubing Material

CHEMICALS	BRASS	ALUMINUM	EPOXY
Nitric Acid 10%	**NR**	**NR**	**R**
Phosphoric Acid 20%	**NR**	**NR**	**R**
Acetic Acid 10%	**NR**	R	**R**
Sodium Hydroxide	Fair	**NR**	**R**
Amonium Hydroxide	**NR**	**NR**	**R**
Calcium Sulfate	R	R	**R**
Sodium Nitrate	R	R	**R**
Sodium Chloride	**NR**	**NR**	**R**
Ammonium Chloride	**NR**	**NR**	**R**
Calcium Hypochloride	**NR**	**NR**	**R**
Water	R	R	**R**
Gasoline	R	R	**R**
Motor Oil	R	R	**R**
Hydraulic Fluid	R	R	**R**

R — Recommended

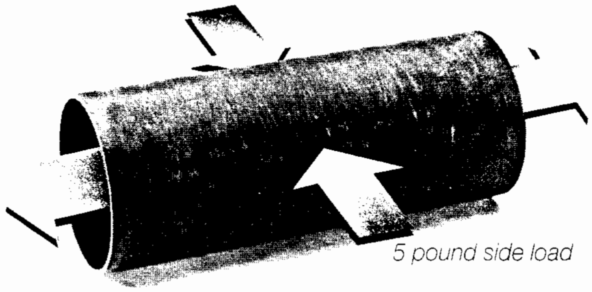
NR — Non-recommended

The list at right is for normal temperatures and operating conditions. For Black-Amalgon's resistance to other chemicals and various temperatures, consult the company for recommendations.

Need More Reasons to Choose Black-Amalgon?

- **No More Catastrophic Failures** — When metal tubing catastrophically fails dangerous shrapnel may result. Should Black-Amalgon fail. there is micro-cracking which simply releases pressure to safe levels.
- **Shape Stability and Impact Resistance** — Ship, store and cut Black-Amalgon — it will retain its circular shape. And unlike metals, the product simply does not dent. Material impact strength is 40 izod ft. lbs.
- **Easy Bonding; Uses Standard End-Fittings** — The material can be easily bonded to most metals using a readily available two-part epoxy bonding kit. (Contact us to discuss bonding to non-metallic materials.) Tie rod or bonded end-cap configurations are the common choice when Black-Amalgon is used as cylinder tubing.
- **Can be Pigmented or Painted** — Need a different color? No problem. Black-Amalgon can be pigmented or painted any color you need. Standard urethane-based paint works well for most applications.
- **Excellent Thermal Stability; Non-Interference** — With a very low coefficient of thermal expansion. Black-Amalgon operates efficiently from -100° F to +270° F. Customers have reported success in using the material in temperatures below -200° F. Unless requested otherwise, the material will not interfere with electronic or magnetic componentry.
- **Machinability** — Black-Amalgon can be cut, chamfered, ground to tight tolerances, shouldered, bored and threaded...to meet your engineering requirements.
- **Best Composite Process** — Black-Amalgon offers the best combination of strength-to-weight ratio, burst strength, and axial loading capability when compared to tubular structures made by other composite processes.

Black-Amalgon
Cycle Tested For Maximum Performance



10.000.000 cycles dry cylinder test without measurable cylinder wear.

Relative Thermal Conductivity of Samples of Black Amalgon In Comparison With Pure Metal Tubes

Material	Density ρ, lbs/in^3	Thermal Conductivity K, BTU/ft•hr•°F
Amalgon	.072	0.250
Brass	.320	61.000
Steel	.280	30.000
Aluminum	.100	132.000
Zinc	.250	65.000
Copper	.320	223.000

Large Inventory of Tooling, Fast Deliveries

Over the years, we have established an extensive inventory of tooling to meet most needs. If not available, we will quote the cost of creating the tooling separately. Currently, tooling is in-house to manufacture Black-Amalgon from 1/2-inch ID to 30-inch ID. Metric sizes are also available. Wall thickness can be varied to meet pressure requirements or component geometry.

If you're in a hurry, lead times are often significantly less than required for metal structures because the self-lubricating, honed-like ID is achieved without lengthy honing. We can also stock products to meet your JIT requirements.

We Go To Any Lengths

Black-Amalgon is sold in random lengths or cut pieces. We will go to any length to provide superior product quality and responsive service.

Founded in 1966, Amalga Composites, Inc. has grown to be leader in designing, engineering and manufacturing filament-wound composites to meet the tough environmental and performance requirements of the fluid power industry.

Interested?

Call for additional technical supplements, sizing and pricing for Black-Amalgon and other composite products available from Amalga Composites, Inc. We're interested in helping you. Call 1-800-262-5424

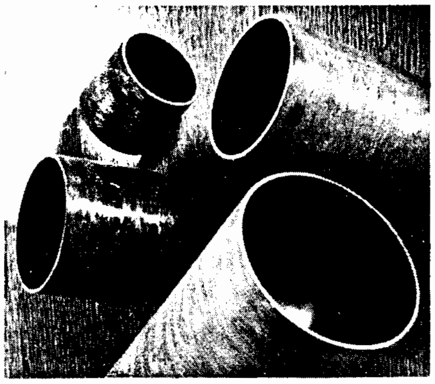
Innovative Composite Structures since 1966

AMALGA COMPOSITES, INC.
10600 West Mitchell Street, West Allis, WI 53214 Phone 414 453-9555 Fax 414 453-9561

PRODUCT SELECTION GUIDE

Black-Amalgon®

Amalga Composites, Inc. ... The Better Choice. **BA.05-93**



lack-Amalgon® is a patented fiber-reinforced thermoset composite material specifically designed for use in cylinder tubing applications. Its material strengths, corrosion resistance, minimum 75% weight savings when compared to steel, and 5-15 RMS microinch bore finish offer significant benefits when compared to traditional metals.

OEM cylinder, actuator and motion control companies, OEM machine builders, end-user systems and equipment manufacturers have used Black-Amalgon® to replace steel, aluminum and brass for years.

More detailed information on strengths and material properties, corrosion resistance, and possible design modifications will be made promptly available upon your request.

STANDARD MEASUREMENT SYSTEM

Model No.[1]	Standard Bore Size (Inches)	Nominal Bore Size (Inches)	Bore Tolerances Available by Selected Procedures (Inches)	Nominal O.D. Size[2] (Inches)	Random Lengths To	Operating PSI Non-Tie Rod Design[3]	Operating PSI Tie Rod Design[3]	Weight Per Foot LBS	Model No.
BA100	1.000	1.000 1.003	−.000 +.002	1.250 1.253	5 feet	1220	2,440	.33	BA100
BA112	1.125	1.125 1.128	−.000 +.002	1.375 1.378	5 feet	1080	2,170	.37	BA112
BA125	1.250	1.250 1.256	−.000 +.002	1.500 1.506	5 feet	980	1,950	.41	BA125
BA150	1.500	1.500 1.506	−.000 +.002	1.750 1.754	5 feet	810	1,620	.50	BA150
BA175	1.750	1.750 1.756	−.000 +.003	2.000 2.004	5 feet	700	1,390	.56	BA175
BA200	2.000	2.000 2.006	−.000 +.003	2.250 2.254	10 feet	610	1,200	.66	BA200
BA225	2.250	2.250 2.258	−.000 +.003	2.500 2.506	10 feet	543	1,000	.75	BA225
BA250	2.500	2.500 2.508	−.000 +.003	2.750 2.760	10 feet	490	970	.83	BA250
BA275	2.750	2.750 2.758	−.000 +.004	3.000 3.010	10 feet	440	890	.91	BA275
BA300	3.000	3.000 3.008	−.000 +.004	3.250 3.260	10 feet	410	810	1.0	BA300
BA325	3.250	3.250 3.258	−.000 +.004	3.500 3.510	10 feet	380	750	1.1	BA325
BA350	3.500	3.500 3.508	−.000 +.005	3.750 3.760	10 feet	350	700	1.2	BA350
BA375	3.750	3.750 3.758	−.000 +.005	4.000 4.010	10 feet	330	650	1.3	BA375
BA400	4.000	4.000 4.010	−.000 +.005	4.250 4.260	10 feet	300	610	1.3	BA400
BA500	5.000	5.000 5.010	−.000 +.005	5.250 5.260	10 feet	240	490	1.7	BA500
BA575	5.750	5.750 5.760	−.000 +.005	6.000 6.012	10 feet	210	420	1.9	BA575
BA600	6.000	6.000 6.010	−.000 +.005	6.250 6.262	10 feet	200	410	2.0	BA600
BA700	7.000	7.000 7.012	−.000 +.007	7.250 7.262	10 feet	170	350	2.3	BA700
BA800	8.000	8.000 8.015	−.000 +.007	8.250 8.265	10 feet	150	300	2.7	BA800
BA1000-A	10.000	10.000 10.020	−.000 +.008	10.370 10.410	10 feet	190	330	5.0	BA1000-A
BA1000-B	10.000	10.000 10.020	−.000 +.008	10.490 10.530	10 feet	260	430	6.7	BA1000-B
BA1200-A	12.000	12.000 12.020	−.000 +.010	12.370 12.410	10 feet	160	280	6.0	BA1200-A
BA1200-B	12.000	12.000 12.020	−.000 +.010	12.490 12.530	10 feet	220	360	8.0	BA1200-B
BA1400-A	14.000	14.000 14.020	−.000 +.010	14.370 14.410	10 feet	140	240	7.0	BA1400-A
BA1400-B	14.000	14.000 14.020	−.000 +.010	14.490 14.530	10 feet	190	310	9.3	BA1400-B
BA1600-B	16.000	16.000 16.025	−.000 +.010	16.490 16.550	10 feet	170	270	10.7	BA1600-B
BA1600-C	16.000	16.000 16.025	−.000 +.010	16.620 16.690	10 feet	210	320	13.4	BA1600-C
BA1800-B	18.000	18.000 18.025	−.000 +.010	18.490 18.550	10 feet	150	240	11.5	BA1800-B

Model No.[1]	Standard Bore Size (Inches)	Nominal Bore Size (Inches)	Bore Tolerances Available by Selected Procedures (Inches)	Nominal O.D. Size[2] (Inches)	Random Lengths To	Operating PSI Non-Tie Rod Design[3]	Operating PSI Tie Rod Design[3]	Weight Per Foot LBS	Model No.
BA1800-C	18.000	18.000 18.025	-.000 +.010	18.620 18.690	10 feet	190	290	15.0	BA1800-C
BA2000-B	20.000	20.000 20.025	-.000 +.010	20.490 20.560	10 feet	130	220	13.5	BA2000-B
BA2000-C	20.000	20.000 20.025	-.000 +.010	20.620 20.700	10 feet	170	260	17.0	BA2000-C
BA2400-C	24.000	24.000 24.025	.000 .020	24.620 24.700	104 inches	140	220	19.1	BA2400-C
BA3000	30.000	30.000 30.025	.000 .020	31.000 31.080	104 inches	190	270	38.6	BA3000

METRIC MEASUREMENT SYSTEM

Model No.[1]	Standard Bore Size (Inches)	Nominal Bore Size (Inches)	Bore Tolerances Available by Selected Procedures (Inches)	Nominal O.D. Size[2] (MM)	Random Lengths To	Operating PSI Non-Tie Rod Design[3]	Operating PSI Tie Rod Design[3]	Weight Per Foot LBS	Model No.
MBA32	32	1.260 1.266	-.000 +.002	38.4	5 Ft.	960	1930	.44	MBA32
MBA40	40	1.575 1.581	-.000 +.002	46.4	5 Ft.	770	1550	.54	MBA40
MBA50	50	1.969 1.975	-.000 +.003	56.4	10 Ft.	610	1240	.65	MBA50
MBA63	63	2.480 2.488	-.000 +.003	69.4	10 Ft.	490	980	.82	MBA63
MBA72	72	2.835 2.843	-.000 +.004	78.4	10 Ft.	454	845	.71	MBA72
MBA80	80	3.150 3.158	-.000 +.004	96.4	10 Ft.	380	770	1.1	MBA80
MBA100	100	3.937 3.947	-.000 +.005	106.4	10 Ft.	310	620	1.3	MBA100
MBA125	125	4.921 4.931	-.000 +.005	121.4	10 Ft.	250	500	1.6	MBA125
MBA125-A	125	4.921 4.931	-.000 +.005	134.5	10 Ft.	400	690	2.4	MBA125-A
MBA160	160	6.299 6.311	-.000 +.005	166.4	10 Ft.	190	390	2.0	MBA160
MBA160-A	160	6.299 6.311	-.000 +.005	169.5	10 Ft.	310	540	3.1	MBA160-A
MBA160-B	160	6.299 6.311	-.000 +.005	172.7	10 Ft.	430	680	4.2	MBA160-B
MBA200	200	7.874 7.889	-.000 +.008	206.4	10 Ft.	150	310	2.5	MBA200
MBA200-A	200	7.874 7.889	-.000 +.008	209.5	10 Ft.	250	430	3.8	MBA200-A
MBA200-B	200	7.874 7.889	-.000 +.008	212.7	10 Ft.	340	550	5.1	MBA200-B

FOOTNOTES: 1. Under 1.000-inch tooling available. Tooling constantly upgraded. Call for availability. 2. Wall thickness can be changed to meet pressure or geometry requirements, from minimum .020 (.5mm) depending upon ID. 3. Operating pressures calculated with minimum 4:1 safety factor.

LET US QUOTE YOUR APPLICATION!

FAX this to 414-453-9561.

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CO.: ______________________

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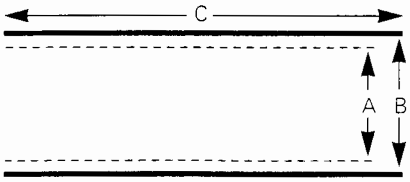

DIMENSIONS

A __________ B __________ C __________

CHAMFER: YES NO ______ DEGREES

SHOULDER: YES NO ______ DEPTH LENGTH __________

HOLES: YES NO ______ DIAMETER LOCATION __________

OTHER OPERATIONS: ______________________ QUANTITY IN PRODUCTION: __________

TOLERANCE REQUIREMENTS: O.D. ______ I.D. ______ LENGTH __________ OTHER ______________________

SAFETY FACTOR: ___ : ___ CYLINDER APPLICATION? YES NO

Innovative Composite Structures Since 1966

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Air-Driven Actuators

The oldest type of air-driven actuator is probably a piston sliding in a cylinder, similar to actuators used for hydraulic systems. For pneumatic operation this type of actuator requires a seal or packing at the piston to keep the leakage at this point down to a minimum. This, of course requires good lubrication. The lubrication necessary for good sealing also reduces friction. Corrosion may be a serious problem for pneumatic actuators and should always be considered when selecting material for these actuators.

Double-Acting Diaphragm Cylinders

Fig. 5–15 is a cutaway view of an air-operated, double-acting diaphragm cylinder. This type of actuator is virtually frictionless and may be designed in many configurations and sizes for different applications. Power loss due to friction is almost negligible, consisting only of the small amount of friction at the piston rod and the minute amount of power required to flex the tough, resilient diaphragm material.

The stroke is long in proportion to the rolling diaphragm height. For instance, a 4-inch (or 10 cm) diameter and 4-inch (or 10 cm) high diaphragm will provide a stroke of about 7 inches (or 18 cm). Similarly, a 6-inch (or 15 cm) diameter and 6 inch (or 15 cm) high diaphragm will provide a total stroke of 10-3/16 inches

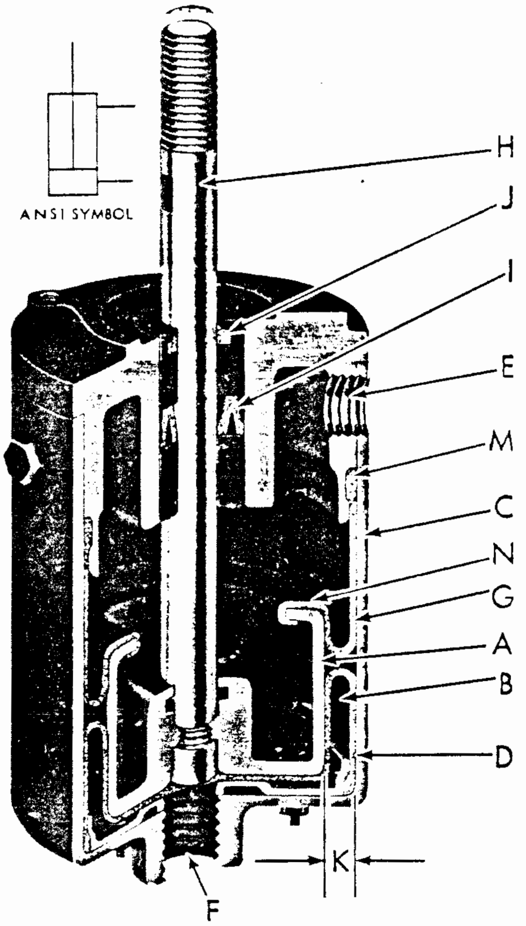


Fig. 5–15. Sectional view of air-operated double-acting diaphragm cylinder. (Courtesy of Bellofram Corporation)

(or 26 cm). The manufacturers claim a working temperature range from −65°F (or −65°C) to 550°F (or 288°C), and in some cases a range of −120°F (or −85°C) to 700°F (or 370°C) is possible.

There are carefully designed diaphragm materials to meet a great variation in applied pressure, ranging from as little as about 2 torr to 500 psi (35 kg/cm²). Because the major portion of the diaphragm carrying the working load is supported by the piston area, the pressure may in some cases be as high as 1,200 psi (85 kg/cm²). Service life is also high, in some cases up to 100 million cycles.

Operation of the cylinder shown in Fig. 5–15 is very simple. Assume that line air pressure is supplied to port *F*. Diaphragm *D* will then roll off piston *A* onto cylinder wall *C*. Diaphragm *G* will roll off cylinder wall *C* onto piston *A*.

According to Pascal's Law, air in a confined space is distributed equally in all directions. Therefore the pressure in space B keeps the piston in a central position within the cylinder without need for bearings to guide it. The only friction, therefore, is the force required to flex the diaphragms plus the small amount of friction between the resilient rod seal and bushing and piston rod *H* as it moves up.

To reverse this action, port *E* previously used as the exhaust port would receive line air pressure and become the inlet port. Port *F*, previously the inlet port, would then be the exhaust port. This reversal is easily done with a four-way valve operated either manually or automatically depending on the application.

With proper valving and controls this type of linear actuator can be made to reciprocate at selected rates of cycling, but in the majority of applications such as positioning, applying pressure, holding and retracting, the time interval between forward and reverse strokes is fairly long and subject to variations. In such cases the actuation is often manually controlled.

As shown in Fig. 5–12, this is a double-acting cylinder where pneumatic pressure moves the piston in both directions, forward and reverse. Therefore the piston rod *H* requires a seal as shown at I to prevent pressure from escaping and also a seal as shown at *J* to keep dust and contamination out. Several designs are available. Some have a spring return, and no air leak whatsoever is possible.

The convolutions, which are in the radial space *K* between cylinder *C* and piston *A* is relatively very small, depending on the diameter of piston and thickness and type of material used for the diaphragm. Standard convolutions range from 1/16" (or 1.5 mm) to 1/4" (6 mm). The sidewall thickness of material ranges from 0.015" (0.4 mm) to 0.035" (0.9 mm). The diaphragms are usually fastened securely in place by providing a bead along the edge as shown at *L* and *M*. In other cases the air pressure will hold the diaphragm in position as shown at *N*.

Because the space *K* is very small, the tension on the side wall of the material is also very small. This tension is easily calculated by assuming the following:

S = stress on material, in pounds per inch of circumference
p = applied pressure, psi
K = width of space in inches

Since the stress S is calculated for 1" of side wall material, the pressure sould be calculated for an area of 1 × K, so that:

$$S = \frac{p \times K \times 1}{2}$$

In the metric system the formula would be the same, with the following values:

S = stress on material in kg per centimeter of circumference
p = applied pressure, kg/cm²
K = width of space in centimeters

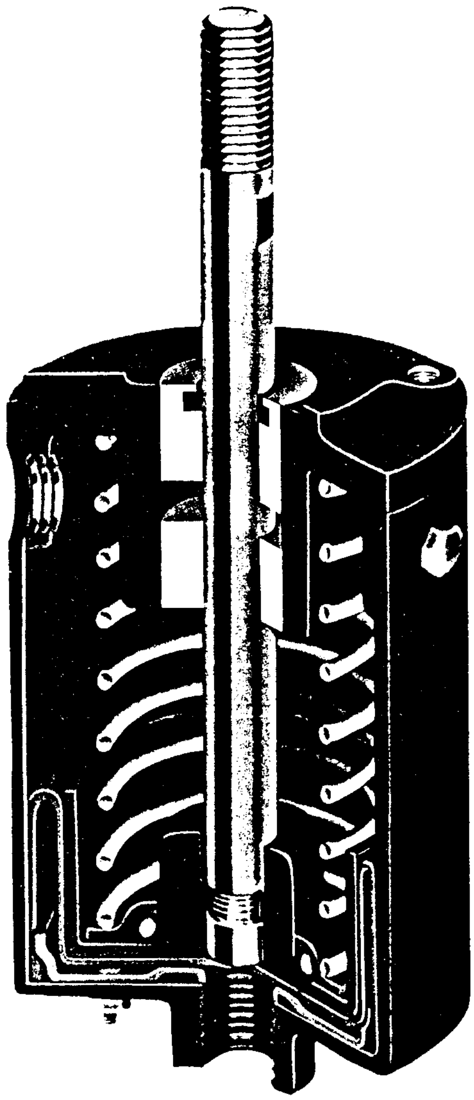
An Introduction to Pneumatic Systems

Anderson

Only very small amounts of leakage can be tolerated in the handling of dangerous fluids such as fluorine. In such cases either bellows or diaphragms are used. A diaphragm has a single involute and is limited in stroke. It also has special problems when the direction of pressure drop changes. Most diaphragms are molded to a shape near their normal operating shape. Consequently, a pressure reversal which causes the convolute to flip also causes excessive stress and shorter life and, of course, discontinuous actuation. The soft materials used in most diaphragms are not able to withstand either very high or very low temperatures. Metal bellows are often used where diaphragm stroke is inadequate or where the temperature causes diaphragm materials to fail.

Bellofram cylinders don't flip over at end of stroke.

"Prime" Mover Catalog



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CORPORATION
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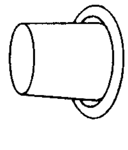
Other Bellofram Products...



Diaphragms
As the leader in the industry we offer diaphragms in 15 standard elastomers and many special compounds which can withstand temperature ranges of −65°F to +600°F and pressures up to 3000 P.S.I.

Bellofram
CORPORATION
a Rexnord Company

The Bellofram Diaphragm Air Cylinder

The development of the long stroke rolling diaphragm for dynamic sealing proved to be the solution for many applications requiring low friction, no lubrication, low leakage, wide temperature variations, and low total cost. The popularity of the rolling diaphragm as a sealing means led to many requests for a standard line of "Off the Shelf" diaphragm cylinders; single and double acting, short and long stroke with a wide selection of effective areas. To meet these requests, the long stroke rolling diaphragm cylinder was developed and Bellofram has supplied many thousands of them since their 1965 introduction.

What are Diaphragm Air Cylinders?

Diaphragm Air Cylinders are actuators made from elastomers, engineered metals and fabrics. They require no lubrication, are virtually frictionless, and economical. They can be used to provide lifting, clamping, pushing, coining, turning, and other linear force or actuation motions in many applications.

Where are they used?

Diaphragm Air Cylinders are replacing conventionally sealed cylinders and actuators where low cost and reliability are requirements. They can be used with vacuum and gaseous pressure systems and applications are almost unlimited. They are currently solving many unique problems, being used as accumulators, pumps, reservoirs, expulsion chambers, shock mounts, impact absorbers, weld drivers, and tensioners.

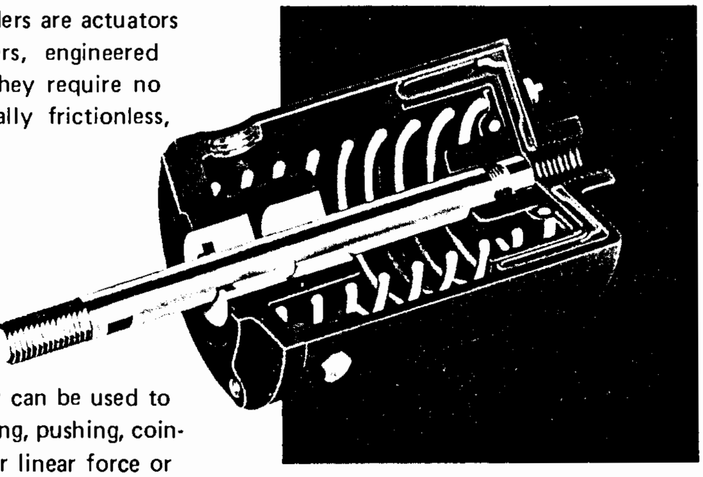
Why use Diaphragm Cylinders?

Use diaphragm cylinders when any or all of the following requirements must be met:

- No Lubrication
- Low Friction
- Extremely Sensitive Response To Small Pressure Variations
- Low Total Cost
- No Blow By Leakage
- Low Start-up Pressure

What materials are used in Diaphragm Air Cylinders?

The diaphragm is constructed of Neoprene® elastomer reinforced with Flex-Weave Dacron® fabric to give exceptionally long cycle life. The cylinder on sizes 4 and 6 is an impact-extruded aluminum shell. Larger sizes are made from a steel shell. The rods are ground, polished and hard-chrome plated steel. The bearings are sintered bronze, molybdenum disulphite impregnated. Other components are high strength materials with suitable corrosion resistant treatment.



Modifications

Our engineering department is continually providing products designed to meet specific customer needs. As a result, a variety of optional components are available, assuring you of the most versatile cylinder. The following are some of the more common options available upon special order:

Springs — A wide variety of special springs are available for any Bellofram air cylinder.

Rods — Rod materials, lengths, and end configurations can be adapted for special applications.

Shells — Special plating or painting of the cylinder shell is available when required.

Diaphragms — The Bellofram diaphragm can be made in an almost unlimited combination of elastomer and fabric. This would include such materials as Nitrile, Silicone, Fluorosilicone, elastomers, and various Dacron® and Nomex® fabrics.

Bearings — In addition to the linear ball bushing, a variety of bearings are available. This would include materials such as Teflon®/glass/Molybdenum (TGM), Celcon® and others.

The major areas for the use of modified cylinders are applications where the cylinder is in contact with corrosive materials, high temperature applications (up to 500°F) and in tension control applications where extreme sensitivity is required.

To date, well over 900 special modified cylinder designs have been produced. It is most likely that we have already modified a cylinder which will meet your needs.

General Operating Information

Bellofram Neoprene Diaphragm Air Cylinders are rated to operate on plant air up to 145 psi (10 bar) over temperatures from −40°F to +225°F.

Special diaphragm materials are available which permit our cylinders to operate at temperatures of −75°F to 400°F.

An air line lubricator is not necessary when operating a Bellofram air cylinder.

It is expected that the installation and operation procedures furnished with each cylinder will be followed for maximum service life.

External stroke limiters should be provided by the customer for limiting the stroke in both directions on single acting as well as double acting cylinders.

Hydraulic or Liquid Service

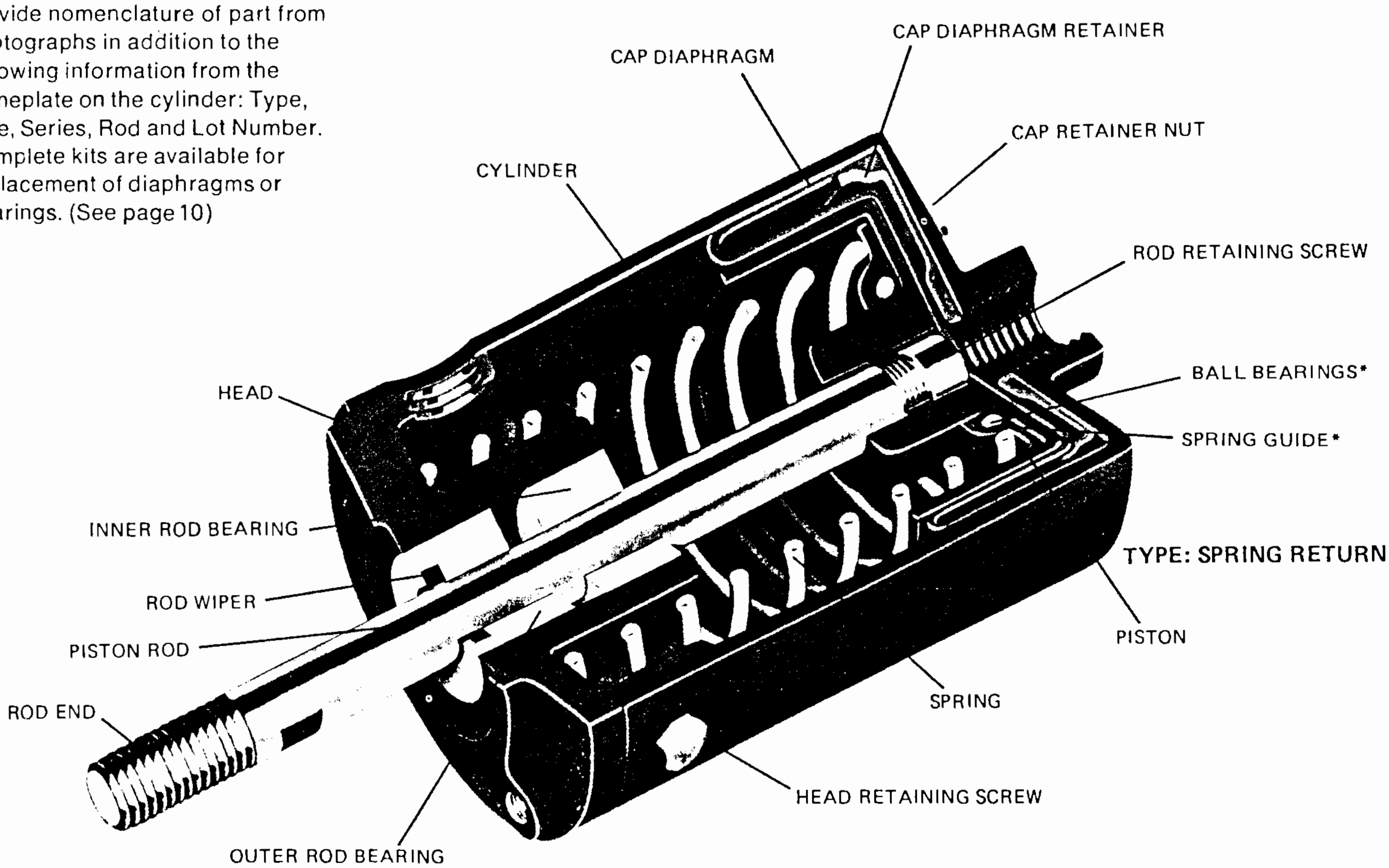
Actuating fluids other than air may be used by simply changing the diaphragm materials. Consult Bellofram's application engineering staff for information on hydraulic or liquid pressurized service.

Testing

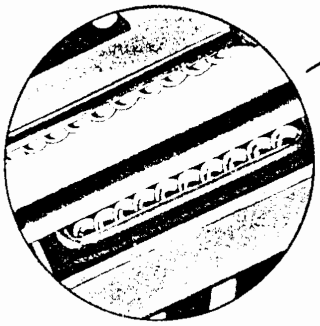
All cylinders are leak-tested prior to shipment. However, the cylinder is not a bubbletight assembly.

Nomenclature

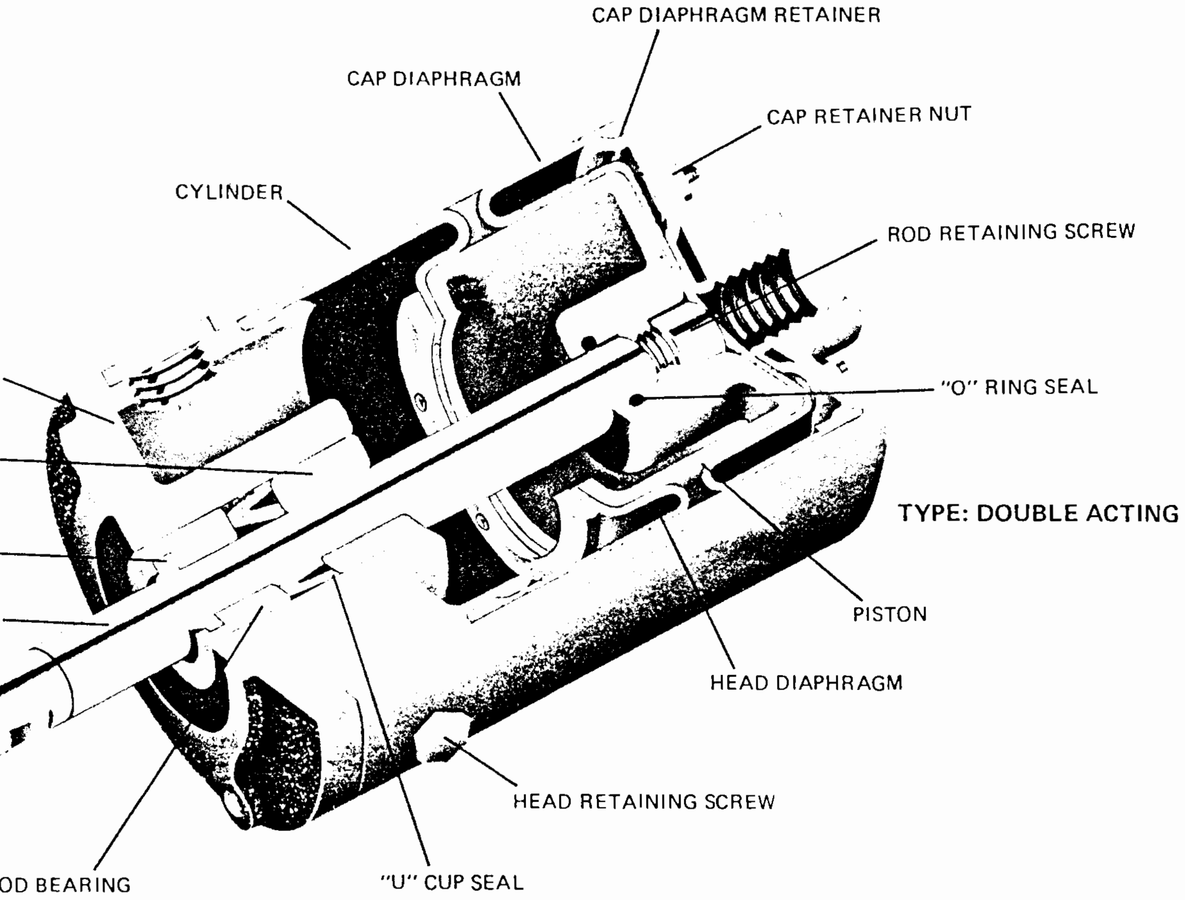
Ordering data for replacement parts
When ordering replacement parts provide nomenclature of part from photographs in addition to the following information from the nameplate on the cylinder: Type, Size, Series, Rod and Lot Number. Complete kits are available for replacement of diaphragms or bearings. (See page 10)



*Supplied on sizes 16F and larger.



Linear Ball Bearings available in Super Cylinder.



Cylinder Weights

Cyl. No.	Lbs.	Cyl. No.	Lbs.	Cyl. No.	Lbs.	Cyl. No.	Lbs.
S-4-F-BP	4	S-16-E-BP	13	D-4-F-BP	4	D-16-E-BP	14
S-4-BP-N	5	S-16-F-BP	14	D-4-N-BP	5	D-16-F-BP	16
S-6-F-BP	5	S-24-E-BP	18	D-6-F-BP	5	D-24-E-BP	20
S-6-BP-N	6	S-24-F-BP	25	D-6-N-BP	6	D-24-F-BP	28
S-9-F-BP	8	S-30-E-BP	25	D-9-F-BP	8	D-30-E-BP	26
S-9-BP-N	9	S-30-F-BP	31	D-9-N-BP	9	D-30-F-BP	33
S-12-E-BP	9	S-36-E-BP	28	D-12-E-BP	10	D-36-E-BP	29
S-12-F-BP	11	S-36-F-BP	36	D-12-F-BP	12	D-36-F-BP	39

Operating Data

Size (Effective Area) Sq. In.	Equiv. Bore Diam. In.	Spring Return						Double Acting	
		Stroke + .03 − .12 Series E In.	Stroke + .03 − .12 Series F In.	Approx. Spring Force — Zero Stroke (lbs.)		Approx. Increase Force Per In. of Stroke (lbs.)		Stroke + .03 − .12 Series E In.	Stroke + .03 − .12 Series F In.
				Series E	Series F	Series E	Series F		
4	2.3		1.80		6		3		1.3
6	2.8		2.40		9		4		1.9
9	3.4	2.20	3.00	17	12	4	4		2.5
12	3.9	2.30	3.60	18	18	6	6	1.8	3.1
16	4.5	2.62	4.20	24	24	8	8	2.1	3.7
24	5.5	2.60	5.24	36	36	11	11	2.0	4.6
30	6.3	3.07	6.00	45	54	13	14	2.4	5.4
36	6.8	3.55	6.00	54	54	16	14	2.9	5.4

Ordering Data

For: Universal, Foot, Clevis Bracket, Clevis Bracket Stud, Head Flange, Cap Flange and Nose Mounts.

D-12-F-BP-HFM

- Type: S — Spring Return, D — Double Acting
- Size: Area in sq. in. 4, 6, 9, 12, 16, 24, 30 or 36
- Series: E or F Letter Determines Stroke
  Make selection from Operating Data Table
- Rod: BP — Bellofram Products Standard.
  SM — National Fluid Power Assoc. Standard
  Make selection from Dimensional Data Table
- Mounting:
  - N — Nose Mount
  - UM — Universal Mount
  - FM — Foot Mount
  - CFM — Cap Flange Mount
  - HFM — Head Flange Mount
  - CBM — Clevis Bracket Mount
  - CBS — Clevis Bracket Stud

Example: D-12-F-BP-HFM is: a double acting, 12 sq. in., 3.1" stroke, BP rod end cylinder with a Head Flange Mount

For: Super Cylinder

All cylinders are "F" stroke single acting only.

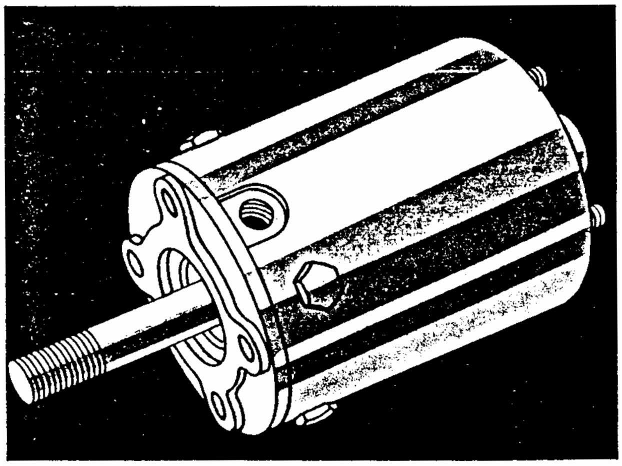
SS-12-F-SM-HFM

- Type: SS — Super Cylinder
- Size: Area in sq. in. 4, 6, 9, 12, 16, 24, 30 or 36
- Series: F Letter Determines Stroke
  Make selection from Operating Data Table
- Rod: BP — Bellofram Products Standard.
  SM — National Fluid Power Assoc. Standard
- Mounting:
  - UM — Universal Mount
  - FM — Foot Mount
  - CFM — Cap Flange Mount
  - HFM — Head Flange Mount
  - CBM — Clevis Bracket Mount
  - CBS — Clevis Bracket Stud

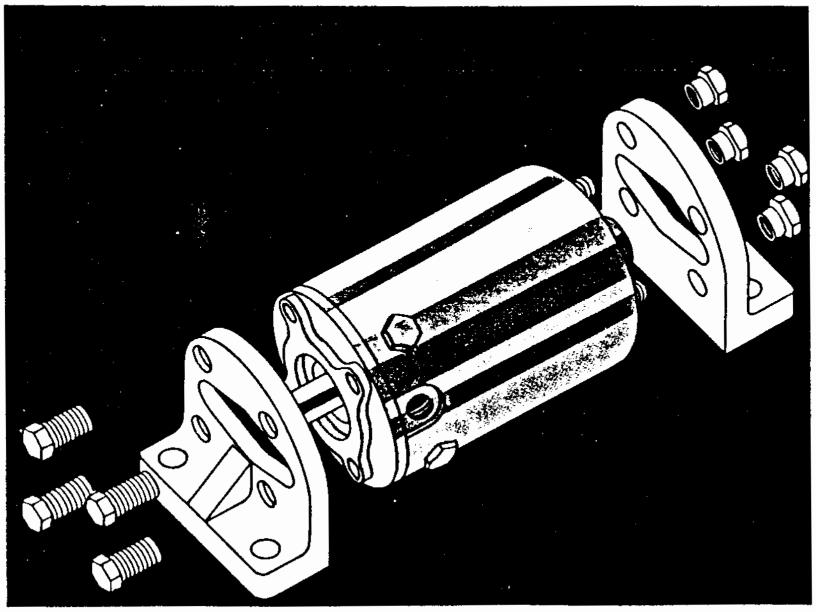
Example: SS-12-F-SM-HFM is: a single acting super cylinder, 12 sq. in., 3.6" stroke, SM rod end cylinder with a Head Flange Mount

STANDARD OPTIONS: NS — No Spring; NB — No Bearing. (Add to end of ordering data.)

Note: CAP MOUNTING STUDS WILL BE FURNISHED ONLY WHEN REQUESTED OR REQUIRED FOR MOUNTING ACCESSORIES ORDERED. Cap mounting stud data and dimensions are described on page 9.



UNIVERSAL MOUNT

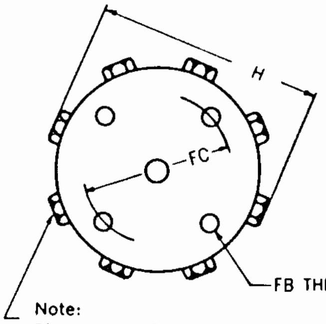


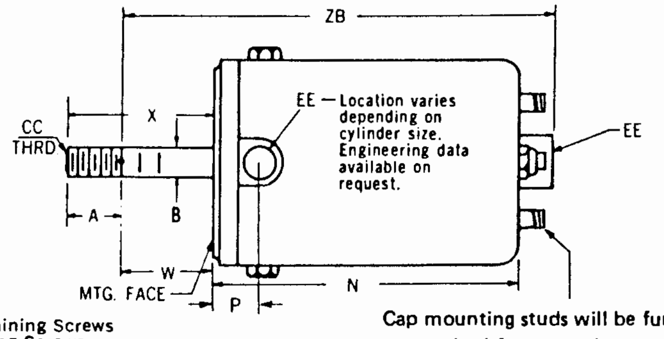
FOOT MOUNT

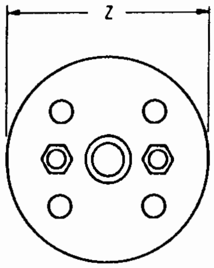
Universal Mount

Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service (10 bar)







Note:
Sizes 4, 6, 9 & 12 have 4 Head Retaining Screws
All other sizes have 8 Head Retaining Screws

Cap mounting studs will be furnished only when requested or required for mounting accessories ordered. See ordering data on page 9.

Dimensional Data — Universal Mount (All dimensions in inches)

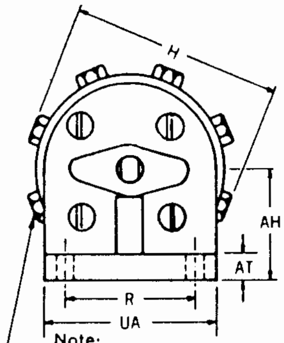
Size	Series	Z	H	N	EE	FC	FB	B	BP Rod End*					SM Rod End†					P
									X	A	W	ZB	CC	X	A	W	ZB	CC	
4	F	2.71	3.02	4.34	¼ NPT	2.00	¼-20	½	2.73	.75	1.98	6.72	⅜-24	2.73	1.00	1.73	6.47	7/16-20	.50
6	F	3.27	3.58	5.28	¼ NPT	2.00	¼-20	½	2.69	.75	1.94	7.63	⅜-24	2.69	1.00	1.69	7.38	7/16-20	.51
9	E	3.84	4.25	5.31	¼ NPT	3.00	7/16-14	¾	2.92	1.00	1.92	7.63	½-20	2.92	1.12	1.80	7.56	¾-16	.75
	F			6.34					2.69		1.69	8.44		2.69		1.57	8.32		
12	E	4.38	4.79	5.31	⅜ NPT	3.00	7/16-14	¾	2.92	1.00	1.92	7.78	½-20	2.92	1.12	1.80	7.66	¾-16	.75
	F			7.28					2.95		1.95	9.78		2.95		1.83	9.66		
16	E	4.99	5.40	6.03	⅜ NPT	3.00	½-13	¾	3.06	1.00	2.06	8.64	½-20	3.06	1.12	1.94	8.52	¾-16	.87
	F			8.38					2.78		1.78	10.71		2.78		1.56	10.59		
24	E	6.16	6.57	6.28	⅜ NPT	4.75	⅝-11	¾	2.86	1.00	1.86	8.73	½-20	2.86	1.12	1.74	8.59	¾-16	1.00
	F			10.22					2.44		1.44	12.08		2.44		1.32	12.03		
30	E	6.88	7.29	7.00	⅜ NPT	4.75	⅝-11	1	2.83	1.25	1.58	9.26	⅝-18	2.83	1.50	1.33	9.30	1-14	1.00
	F			11.44					3.05	1.50	1.55	13.53	1-12	3.05		1.55	13.53		
36	E	7.38	7.79	7.69	⅜ NPT	4.75	⅝-11	1	2.83	1.25	1.58	9.82	⅝-18	2.83	1.50	1.33	10.00	1-14	1.00
	F			11.47					3.05	1.50	1.55	13.54	1-12	3.05		1.55	13.54		

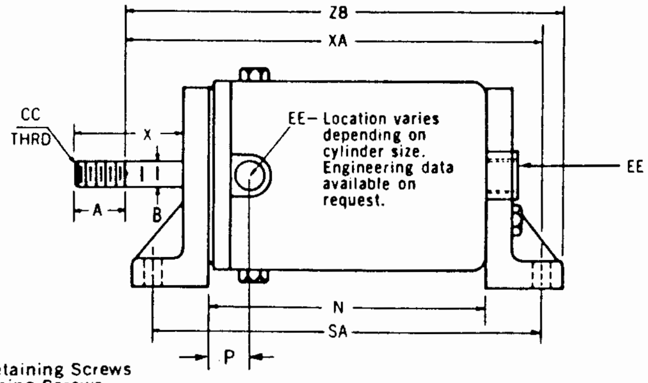
*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards

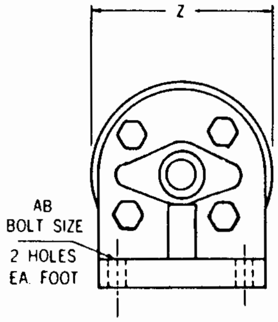
Foot Mount

Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service (10 bar)







Note:
Sizes 4, 6, 9 & 12 have 4 Head Retaining Screws
All other sizes have 8 Head Retaining Screws

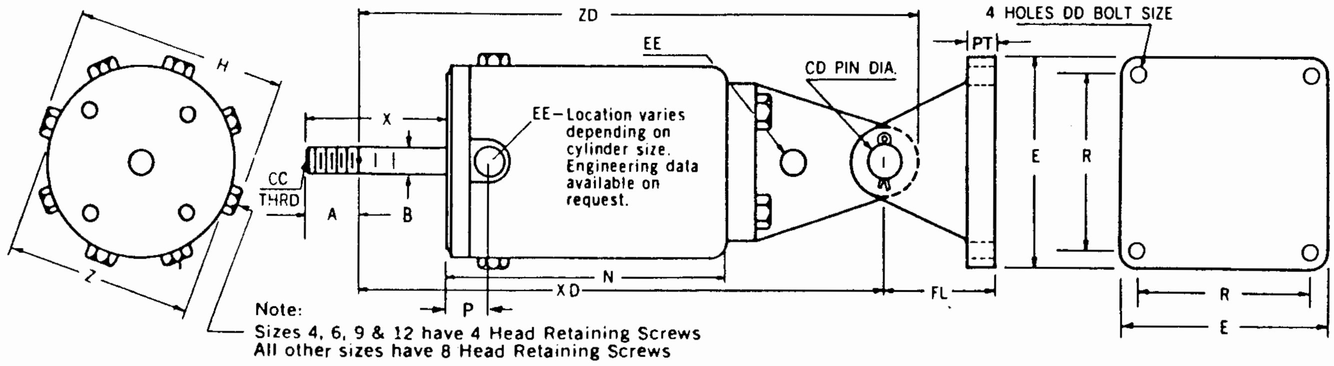
Dimensional Data — Foot Mount (All dimensions in inches)

Size	Series	Z	H	N	EE	B	BP Rod End*					SM Rod End†					AT	AH	UA	R	SA	AB	P
							X	A	XA	ZB	CC	X	A	XA	ZB	CC							
4	F	2.71	3.02	4.34	¼ NPT	½	2.41	.75	7.44	7.82	⅜-24	2.41	1.00	7.19	7.57	7/16-20	.38	1.88	2.62	2.00	6.59	¼	.50
6	F	3.27	3.58	5.28	¼ NPT	½	2.38	.75	8.35	8.72	⅜-24	2.38	1.00	8.10	8.47	7/16-20	.38	1.88	2.62	2.00	7.53	¼	.51
9	E	3.84	4.15	5.31	¼ NPT	¾	2.24	1.00	8.86	9.55	½-20	2.24	1.12	8.74	9.43	¾-16	.56	2.75	4.00	3.00	8.56	½	.75
	F			6.34			2.06		9.67	10.29		2.06		9.55	10.16						9.59		
12	E	4.38	4.79	5.31	⅜ NPT	¾	2.30	1.00	8.86	9.56	½-20	2.30	1.12	8.73	9.44	¾-16	.56	2.75	4.00	3.00	8.56	½	.75
	F			7.28			2.33		10.86	11.56		2.33		10.73	11.36						10.53		
18	E	4.99	5.40	6.03	⅜ NPT	¾	2.31	1.00	9.72	10.42	½-20	2.34	1.12	9.59	10.22	¾-16	.58	2.75	4.00	3.00	9.28	½	.87
	F			8.38			2.16		11.78	12.41		2.16		11.66	12.28						11.62		
24	E	6.16	6.57	6.28	⅜ NPT	¾	2.23	1.00	10.09	10.86	½-20	2.23	1.12	9.94	10.63	¾-16	.68	4.00	6.25	4.75	10.16	⅝	1.00
	F			10.22			1.80		13.06	14.29		1.80		13.10	13.79						14.09		
30	E	6.88	7.29	7.00	⅜ NPT	1	2.20	1.25	10.52	11.20	⅝-18	2.20	1.50	10.26	10.95	1-14	.68	4.00	6.25	4.75	10.88	⅝	1.00
	F			11.44			2.41	1.50	14.92	15.61	1-12	2.41		14.92	15.69						15.31		
36	E	7.38	7.79	7.69	⅜ NPT	1	2.20	1.25	11.22	11.91	⅝-18	2.20	1.50	10.97	11.66	1-14	.68	4.00	6.25	4.75	11.56	⅝	1.00
	F			11.47			2.41	1.50	14.94	15.62	1-12	2.41		14.94	15.62						15.34		

*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards

Clevis Bracket Mount (or Stud)** Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service (10 bar)



Dimensional Data — Clevis Bracket Mount (All dimensions in inches)

Size	Series	Z	H	N	EE	B	BP Rod End*					SM Rod End†					CD	DD	R	E	FL	EW	PT	P
							X	A	XD	ZD	CC	X	A	XD	ZD	CC								
4	F	2.71	3.02	4.34	¼ NPT	½	2.73	.75	8.45	9.07	⅜-24	2.73	1.00	8.19	8.82	7/16-20	.625	¼	2.38	3.12	1.38	.93	⅜	.50
6	F	3.27	3.58	5.28	¼ NPT	½	2.69	.75	9.35	9.97	⅜-24	2.69	1.00	9.09	9.72	7/16-20	.625	¼	2.38	3.12	1.38	.93	½	.51
9	E	3.84	4.25	5.31	¼ NPT	¾	2.92	1.00	9.98	10.73	½-20	2.92	1.12	9.86	10.61	¾-16	.750	¼	3.00	3.75	1.69	.99	½	.75
	F			6.34			2.69		10.80	11.55		2.69		10.67	11.42									
12	E	4.38	4.79	5.31	⅜ NPT	¾	2.92	1.00	10.23	10.98	½-20	2.92	1.12	10.11	10.86	¾-16	.750	⅜	3.00	4.00	1.75	1.24	½	.75
	F			7.28			2.95		12.23	12.98		2.95		12.11	12.86									
16	E	4.99	5.40	6.03	⅜ NPT	¾	3.06	1.00	11.09	11.84	½-20	3.06	1.12	10.97	11.72	¾-16	.750	½	3.00	4.00	1.75	1.24	½	.87
	F			8.38			2.78		13.16	13.91		2.78		13.03	13.78									
24	E	6.16	6.57	6.28	⅜ NPT	¾	2.86	1.00	10.78	11.78	½-20	2.86	1.12	11.16	12.16	¾-16	1.000	½	4.00	5.12	2.00	1.49	⅝	1.00
	F			10.22			2.44		15.22	16.22		2.44		14.68	15.68									
30	E	6.88	7.29	7.00	⅜ NPT	1	2.83	1.25	11.70	12.70	⅝-18	2.83	1.50	11.89	12.89	1-14	1.000	½	4.00	5.12	2.00	1.49	⅝	1.00
	F			11.44			3.05	1.50	16.11	17.11	1-12	3.05		16.11	17.11									
36	E	7.38	7.79	7.69	⅜ NPT	1	2.83	1.25	12.31	13.31	⅝-18	2.83	1.50	12.60	13.60	1-14	1.000	½	4.00	5.12	2.00	1.49	⅝	1.00
	F			11.47			3.05	1.50	16.13	17.13	1-12	3.05		16.13	17.13									

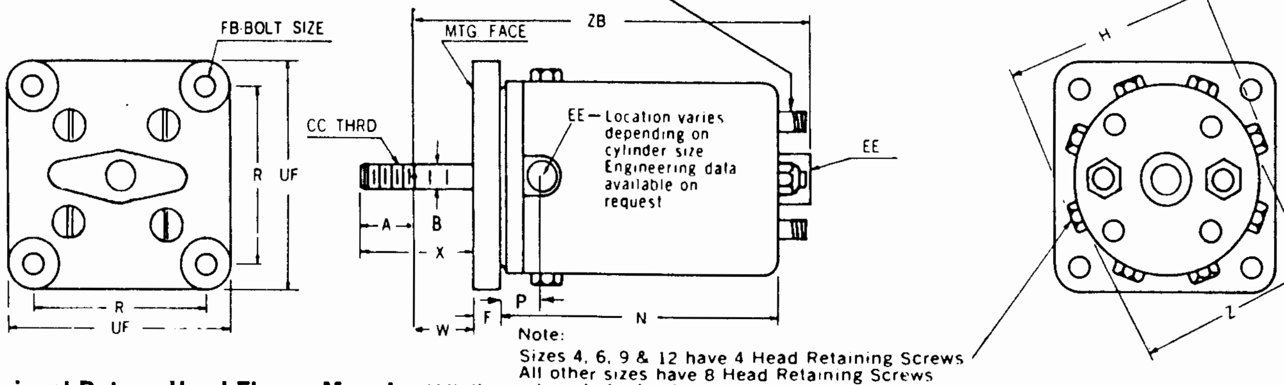
* BP Rod End — Bellofram Products Co. Standard
** See note under Clevis Bracket Mount illustration on next page
† SM Rod End — National Fluid Power Assoc. Standards

Head Flange Mount Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service (10 bar)

Cap mounting stud data and dimensions available on request.

Cap mounting studs will be furnished only when requested or required for mounting accessories ordered.



Dimensional Data — Head Flange Mount (All dimensions in inches)

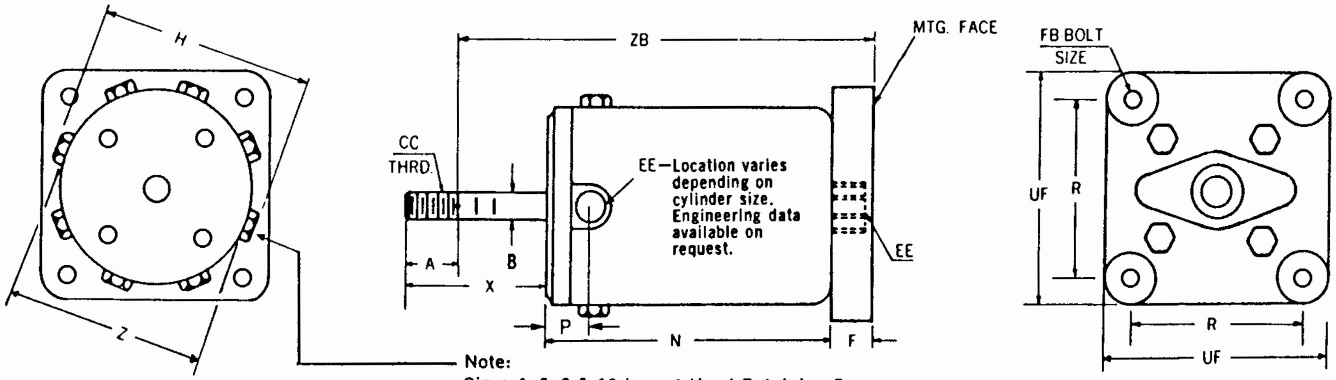
Size	Series	Z	H	N	EE	B	BP Rod End*					SM Rod End†					F	R	UF	FB	P
							X	A	W	ZB	CC	X	A	W	ZB	CC					
4	F	2.71	3.02	4.34	¼ NPT	½	1.95	.75	1.20	6.72	⅜-24	1.95	1.00	0.95	6.47	7/16-20	.781	2.81	3.62	¼	.50
6	F	3.27	3.58	5.28	¼ NPT	½	1.91	.75	1.16	7.63	⅜-24	1.91	1.00	0.91	7.38	7/16-20	.781	2.81	3.62	¼	.51
9	E	3.84	4.15	5.31	¼ NPT	¾	2.23	1.00	1.23	7.63	½-20	2.23	1.12	1.11	7.56	¾-16	.690	4.38	5.50	5/16	.75
	F			6.34			2.00		1.00	8.44		2.00		0.88	8.32						
12	E	4.38	4.79	5.31	⅜ NPT	¾	2.23	1.00	1.23	7.78	½-20	2.23	1.12	1.11	7.66	¾-16	.690	4.38	5.50	7/16	.75
	F			7.28			2.26		1.26	9.78		2.26		1.14	9.66						
16	E	4.99	5.40	6.03	⅜ NPT	¾	2.37	1.00	1.36	8.64	½-20	2.37	1.12	1.25	8.52	¾-16	.690	4.38	5.50	½	.87
	F			8.38			2.09		1.09	10.71		2.09		0.97	10.59						
24	E	6.16	6.57	6.28	⅜ NPT	¾	2.17	1.00	1.17	8.73	½-20	2.17	1.12	1.05	8.59	¾-16	.656	6.00	7.50	½	1,00
	F			10.22			1.78		0.78	12.08		1.75		0.63	12.03						
30	E	6.88	7.29	7.00	⅜ NPT	1	2.14	1.25	0.99	9.26	⅝-18	2.14	1.50	0.64	9.30	1-14	.656	6.00	7.50	½	1.00
	F			11.44			2.36	1.50	0.86	13.53	1-12	2.36		0.86	13.53						
36	E	7.38	7.79	7.69	⅜ NPT	1	2.14	1.25	0.99	9.82	⅝-18	2.14	1.50	0.64	10.00	1-14	.656	6.00	7.50	½	1.00
	F			11.47			2.36	1.50	0.86	13.54	1-12	2.36		0.86	13.69						

*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards

Cap Flange Mount

Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service (10 bar)

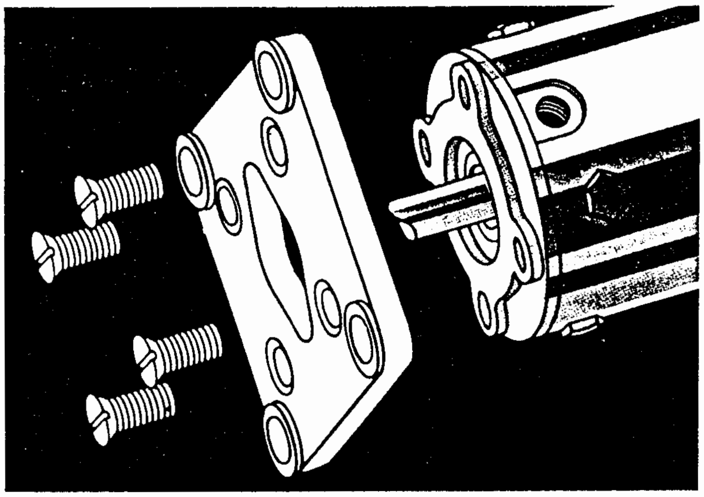


Dimensional Data — Cap Flange Mount (All dimensions in inches)

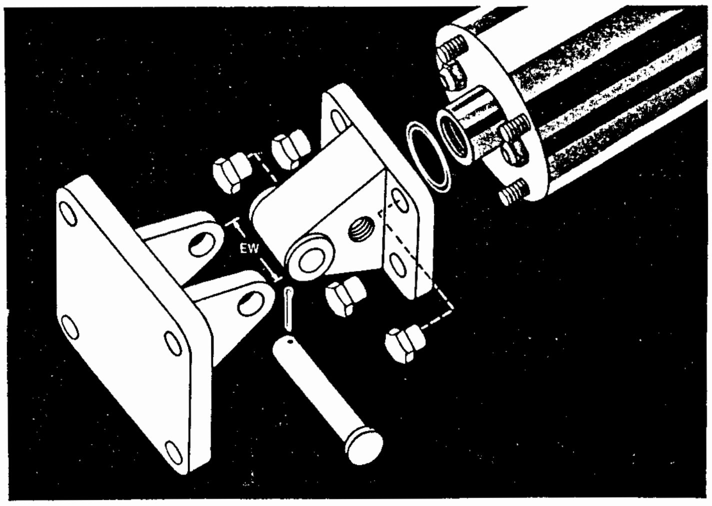
Size	Series	Z	H	N	EE	B	BP Rod End*				SM Rod End†				F	R	UF	FB	P
							X	A	ZB	CC	X	A	ZB	CC					
4	F	2.71	3.02	4.34	¼ NPT	½	2.73	.75	7.10	⅜-24	2.73	1.00	6.85	7/16-20	.781	2.81	3.62	¼	.50
6	F	3.27	3.58	5.28	¼ NPT	½	2.69	.75	8.00	⅜-24	2.69	1.00	7.75	7/16-20	.781	2.81	3.62	¼	.51
9	E	3.84	4.15	5.31	¼ NPT	¾	2.92	1.00	7.92	½-20	2.92	1.12	7.80	¾-16	.690	4.38	5.50	7/16	.75
	F			6.34			2.69		8.72		2.69		8.60						
12	E	4.38	4.79	5.31	⅜ NPT	¾	2.92	1.00	7.92	½-20	2.92	1.12	7.80	¾-16	.690	4.38	5.50	7/16	.75
	F			7.28			2.95		9.92		2.95		9.80						
16	E	4.99	5.40	6.03	⅜ NPT	¾	3.06	1.00	8.18	½-20	3.06	1.12	8.66	¾-16	.690	4.38	5.50	½	.87
	F			8.38			2.78		10.85		2.78		10.73						
24	E	6.16	6.57	6.28	⅜ NPT	¾	2.86	1.00	8.53	½-20	2.86	1.12	8.71	¾-16	.690	6.00	7.50	½	1.00
	F			10.22			2.44		12.35		2.44		12.23						
30	E	6.88	7.29	7.00	⅜ NPT	1	2.83	1.25	9.27	⅝-18	2.83	1.50	8.02	1-14	.690	6.00	7.50	½	1.00
	F			11.44			3.05	1.50	13.68	1.12	3.05		13.68						
36	E	7.38	7.79	7.69	⅜ NPT	1	2.83	1.25	9.96	⅝-18	2.83	1.50	9.71	1-14	.690	6.00	7.50	½	1.00
	F			11.47			3.05	1.50	13.71	1.12	3.05	1.50	13.71						

*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards

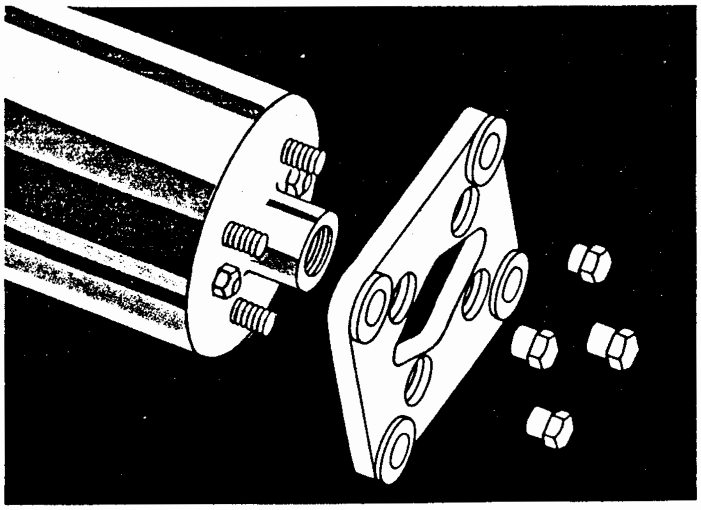
HEAD FLANGE MOUNT



CLEVIS BRACKET MOUNT



Clevis Bracket Mount includes parts as shown in white. Clevis Bracket Stud includes only the Male Bracket, O-Ring and four Sleeve Nuts. Note: Refer to top of opposite page for "EW" dimensions.

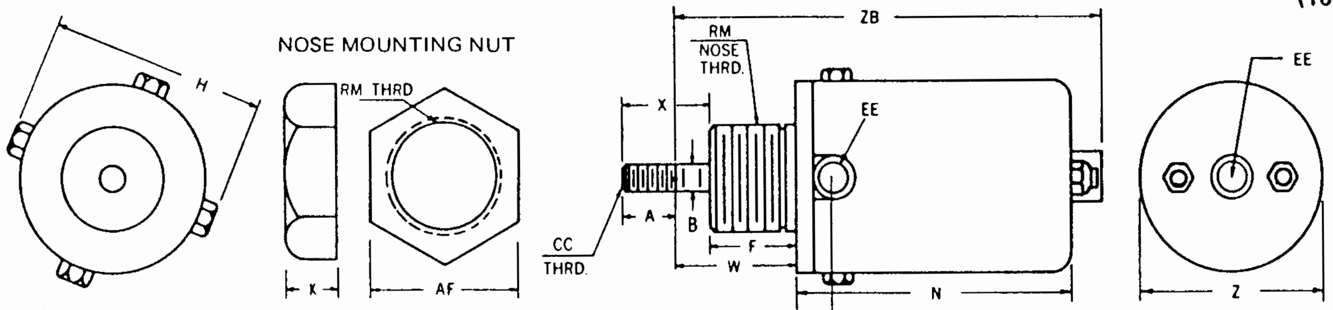


CAP FLANGE MOUNT

Nose Mount

Type: Spring Return or Double Acting, Stroke: Series E or F

145 psi service
(10 bar)



Dimensional Data — Nose Mount (All dimensions in inches)

Size	Series	Z	H	N	EE	F	RM	B	BP Rod End* X	BP Rod End* A	BP Rod End* W	BP Rod End* ZB	BP Rod End* CC	SM Rod End† X	SM Rod End† A	SM Rod End† W	SM Rod End† ZB	SM Rod End† CC	AF	K	P
4	F	2.71	3.02	4.34	¼ NPT	1.25	1⅜-12	½	1.48	.75	1.98	6.72	⅜-24	1.48	1.00	1.73	6.47	7/16-20	2.06	.78	.59
6	F	3.27	3.58	5.28	¼ NPT	1.25	1⅜-12	½	1.44	.75	1.94	7.63	⅜-24	1.44	1.00	1.69	7.06	7/16-20	2.06	.78	.59
9	E	3.84	4.15	5.16	¼ NPT	1.25	1⅝-12	¾	1.83	1.00	2.08	7.65	½-20	1.83	1.12	1.96	7.53	¾-16	2.44	.91	.59
	F			6.19					1.61		1.86	8.45		1.61		1.74	8.33				

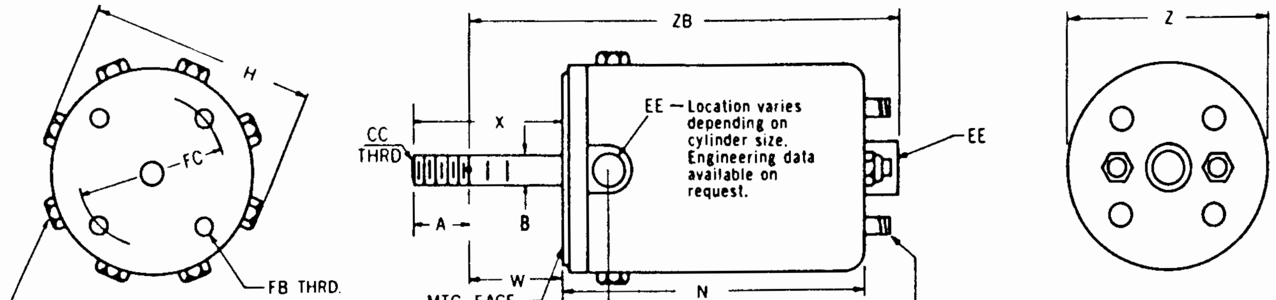
*BP Rod End — Bellofram Products Co. Standard †SM Rod End — National Fluid Power Assoc. Standards

Super Cylinder

145 psi service
(10 bar)

For applications where a maximum of sensitivity is required, we now offer the "Super Cylinder".

The super sensitivity of this model is made possible by the incorporation of a linear ball bearing and hardened steel rod. Friction is reduced to an absolute minimum, making the Super Cylinder virtually frictionless.

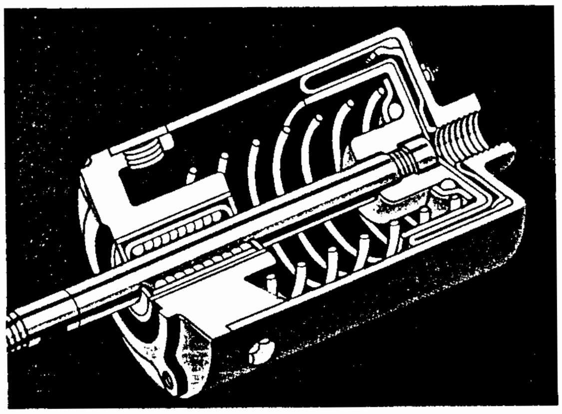


Note:
Sizes 4, 6, 9 & 12 have 4 Head Retaining Screws
All other sizes have 8 Head Retaining Screws

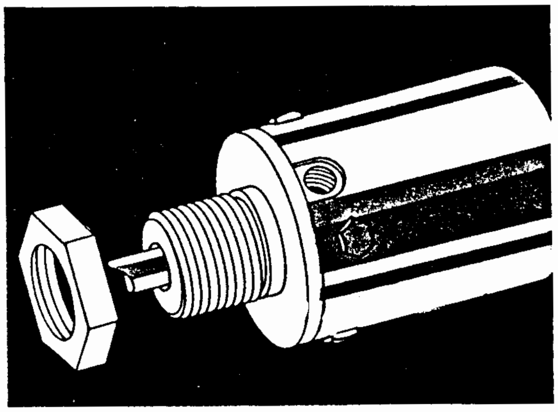
Cap mounting studs will be furnished only when requested or required for mounting accessories ordered. See ordering data, next page.

Dimensional Data — Super Cylinder (All dimensions in inches)

Size	Z	H	N	EE	FC	FB	B	BP Rod End X	BP Rod End A	BP Rod End W	BP Rod End ZB	BP Rod End CC	SM Rod End X	SM Rod End A	SM Rod End W	SM Rod End ZB	SM Rod End CC	P
4	2.71	3.02	4.34	1/4 NPT	2.00	1/4-20	1/2	3.10	.75	2.35	7.09	3/8-24	3.10	1.00	2.10	6.85	7/16-20	.50
6	3.27	3.58	5.28	1/4 NPT	2.00	1/4-20	1/2	2.16	.75	1.41	7.09	3/8-24	2.16	1.00	1.16	6.85	7/16-20	.51
9	3.84	4.25	6.34	1/4 NPT	3.00	7/16-14	3/4	3.50	1.00	2.50	9.25	1/2-20	3.50	1.12	2.38	9.13	3/4-16	.75
12	4.38	4.79	7.28	3/8 NPT	3.00	7/16-14	3/4	2.57	1.00	1.57	9.38	1/2-20	2.57	1.12	1.45	9.26	3/4-16	.75
16	4.99	5.40	8.38	3/8 NPT	3.00	1/2-13	3/4	3.78	1.00	2.78	11.69	1/2-20	3.78	1.12	2.66	11.57	3/4-16	.87
24	6.16	6.57	10.22	3/8 NPT	4.75	5/8-11	3/4	2.00	1.00	1.00	11.75	1/2-20	2.00	1.12	1.00	11.75	3/4-16	1.00
30	6.88	7.29	11.44	3/8 NPT	4.75	5/8-11	1	3.05	1.50	1.55	13.52	1-12	3.05	1.50	1.55	13.52	1-14	1.00
36	7.38	7.79	11.47	3/8 NPT	4.75	5/8-11	1	3.05	1.50	1.55	13.55	1-12	3.05	1.50	1.55	13.55	1-14	1.00



SUPER CYLINDER

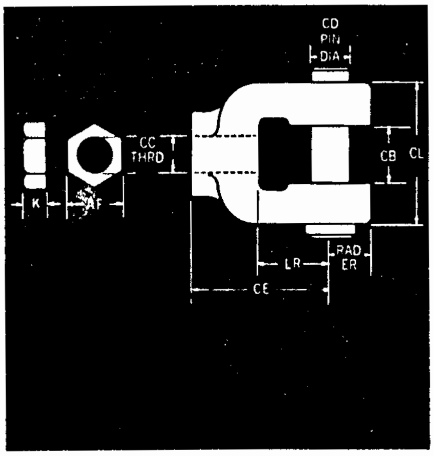


NOSE MOUNT

Rod Clevis and Jam Nut

Size	BP* Rod End CC Thrd.	SM† Rod End CC Thrd.	Series	CB	CD Pin Dia.	CE	CL	ER Rad.	LR	AF	K
4	3/8-24	–	F	.56	3/8	1 7/8	1.38	.53	1.25	9/16	7/32
	–	7/16-20	F	.56	7/16	1 7/8	1.38	.53	1.25	11/16	1/4
6	3/8-24	–	F	.56	3/8	1 7/8	1.38	.53	1.25	9/16	7/32
	–	7/16-20	F	.56	7/16	1 7/8	1.38	.53	1.25	11/16	1/4
9	1/2-20	–	E	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	1/2-20	–	F	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	–	3/4-16	E	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
	–	3/4-16	F	.83	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
12	1/2-20	–	E	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	1/2-20	–	F	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	–	3/4-16	E	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
	–	3/4-16	F	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
16	1/2-20	–	E	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	1/2-20	–	F	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	–	3/4-16	E	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
	–	3/4-16	F	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
24	1/2-20	–	E	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	1/2-20	–	F	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	–	3/4-16	E	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
	–	3/4-16	F	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
30	5/8-18	–	E	.56	9/16	1 7/8	1.38	.53	1.25	15/16	3/8
	1-12	–	F	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
	–	1-14	E	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
	–	1-14	F	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
36	5/8-18	–	E	.56	9/16	1 7/8	1.38	.53	1.25	15/16	3/8
	1-12	–	F	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
	–	1-14	E	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
	–	1-14	F	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64

*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards



Ordering Data

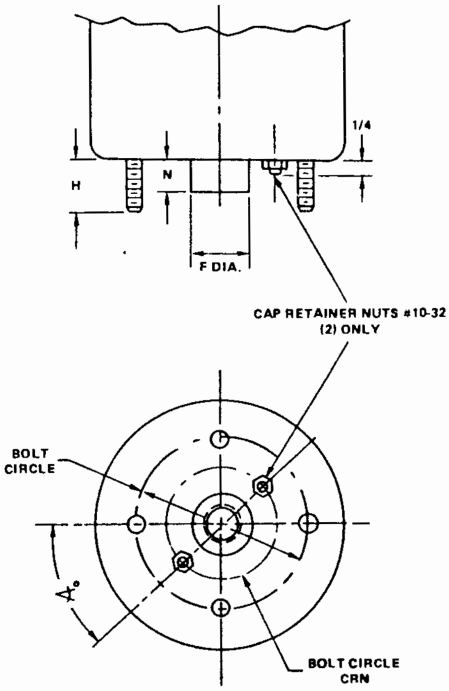
36-F-BP-RC

- Size: Area in sq. in. 4, 6, 9, 12, 16, 24, 30 or 36 } From cylinder on which clevis is to be used.
- Series: E or F Letter Determines Stroke. Make selection from Operating Data Table } From cylinder on which clevis is to be used.
- Rod Clevis
- Rod: BP — Bellofram Products Standard; SM — National Fluid Power Assn. Standard. Make selection from Dimensional Data Table

Example: 36-F-BP-RC is a Rod Clevis for a size 36 sq. in. cylinder with a 1-12 rod thread.

Cylinder Cap End Mounting Studs

Size	No. of Studs	Size of Stud	H (Approx.)	Bolt Circle	F (Boss)	N	Bolt Circle CRN	∢°
4	2	1/4-20	5/8	1-5/16	11/16	13/32	1-3/8	90
6	2	1/4-20	5/8	1-1/2	11/16	13/32	1-3/8	90
9	4	1/4-20	5/8	2	11/16	13/32	1-3/8	45
12	4	1/4-20	5/8	2-5/16	1	17/32	1-11/16	45
16	4	3/8-16	5/8	2-5/16	1	17/32	1-11/16	45
24	4	3/8-16	9/16	3-1/8	1	17/32	1-11/16	45
30	4	1/2-13	11/16	4	1	17/32	1-11/16	45
36	4	1/2-13	11/16	4	1	17/32	1-11/16	45



Repair Kits

The following is included in the repair kits:

Spring Return Diaphragm Kit

1. Diaphragm, Cap
2. Retainer, Adhesive, Cap
3. Nuts, Cap Retainer
4. Instructions

Spring Return Bearing Kit

1. Inner Bearing
2. Outer Bearing
3. Rod Wiper
4. Instructions

Double Acting Diaphragm Kit

1. Diaphragm, Cap
2. Diaphragm, Head
3. Retainer, Adhesive, Cap
4. Retainer, Adhesive, Head
5. Rivets, Blind (or screws)
6. Nuts, Cap Retainer
7. Seal "O" Ring
8. Instructions

Double Acting Bearing Kit

1. Inner Bearing
2 Outer Bearing
3. Rod Wiper
4. U-Cup Seal
5. Instructions

Repair Kits are available to permit user in-plant maintenance without delay and expense of returning parts to the factory. Each kit includes installation instructions. Nameplate data of the cylinder must accompany order to insure receipt of correct parts.

Kits For Spring Return Cylinders

For Models	Diaphragm Kit No.
S-4-F-BP S-4-F-BP-N S-4-F-SM S-4-F-SM-N	S4FN
S-6-F-BP S-6-F-BP-N S-6-F-SM S-6-F-SM-N	S6FN
S-9-E-BP S-9-E-BP-N S-9-E-SM S-9-E-SM-N	S9EN
S-9-F-BP S-9-F-BP-N S-9-F-SM S-9-F-SM-N	S9FN
S-12-E-BP S-12-E-SM	S12E
S-12-F-BP S-12-F-SM	S12F
S-16-E-BP S-16-E-SM	S16E
S-16-F-BP S-16-F-SM	S16F
S-24-E-BP S-24-E-SM	S24E
S-24-F-BP S-24-F-SM	S24F
S-30-E-BP S-30-E-SM	S30E
S-30-F-BP S-30-F-SM	S30F
S-36-E-BP S-36-E-SM	S36E
S-36-F-BP S-36-F-SM	S36F

For Models	Bearing Kit No.
S-4-F-BP S-4-F-BP-N S-4-F-SM S-4-F-SM-N S-6-F-BP S-6-F-BP-N S-6-F-SM S-6-F-SM-N	SB46S
S-9-E-BP S-9-E-BP-N S-9-E-SM S-9-E-SM-N S-9-F-BP S-9-F-BP-N S-9-F-SM S-9-F-SM-N S-12-E-BP S-12-E-SM S-12-F-BP S-12-F-SM S-16-E-BP S-16-E-SM S-16-F-BP S-16-F-SM S-24-E-BP S-24-E-SM S-24-F-BP S-24-F-SM	SB924S
S-30-E-BP S-30-E-SM S-30-F-BP S-30-F-SM S-36-E-BP S-36-E-SM S-36-F-BP S-36-F-SM	SB36FB

Kits For Double Acting Cylinders

For Models	Diaphragm Kit No.
D-4-F-BP D-4-F-BP-N D-4-F-SM D-4-F-SM-N	D4S
D-6-F-BP D-6-F-BP-N D-6-F-SM D-6-F-SM-N	D6S
D-9-F-BP D-9-F-BP-N D-9-F-SM D-9-F-SM-N	D9S
D-12-E-BP D-12-E-SM	D12ES
D-12-F-BP D-12-F-SM	D12FS
D-16-E-BP D-16-E-SM	D16ES
D-16-F-BP D-16-F-SM	D16FS
D-24-E-BP D-24-E-SM	D24ES
D-24-F-BP D-24-F-SM	D24FS
D-30-E-BP D-30-E-SM	D30ES
D-30-F-BP D-30-F-SM	D30FS
D-36-E-BP D-36-E-SM	D36ES
D-36-F-BP D-36-F-SM	D36FS

For Models	Bearing Kit No.
D-4-F-BP D-4-F-BP-N D-4-F-SM D-4-F-SM-N D-6-F-BP D-6-F-BP-N D-6-F-SM D-6-F-SM-N	DB46S
D-9-F-BP D-9-F-BP-N D-9-F-SM D-9-F-SM-N D-12-E-BP D-12-E-SM D-12-F-BP D-12-F-SM D-16-E-BP D-16-E-SM D-16-F-BP D-16-F-SM D-24-E-BP D-24-E-SM D-24-F-BP D-24-F-SM	DB924S
D-30-E-BP D-30-E-SM D-30-F-BP D-30-F-SM D-36-E-BP D-36-E-SM D-36-F-BP D-36-F-SM	DB36FB

Ordering Data
Order by kit number above

Breather Vents

Breather vents are available for use on Bellofram Spring Return Air Cylinders.

The Breather, which contains a 40 micron bronze filter, is simply threaded into the air relief port of the cylinder head. It prevents foreign matter from being drawn into the the cylinder on the return stroke of the piston, and also acts as a snubber. The snubbing reduces the piston speed and impact at the end of the stroke in both directions.

Ordering Data
Breather Vent for ¼" pipe tap
(Fits cylinder sizes -4, -6, -9)
Part No. 315-661-001
(Formerly #BV-2)

Breather Vent for ⅜" pipe tap
(Fits cylinder sizes -12, -16, -24, -30, -36)
Part No. 315-661-002
(Formerly #BV-3)

Warranty

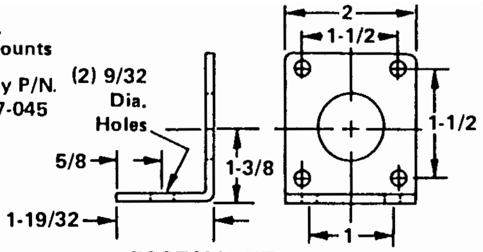
Bellofram Corporation guarantees the products of its manufacture to be free of defects of materials and workmanship in normal use for a period of ninety (90) days from date of sale to customer. The guarantee is limited to repair or replacement of the defective product at the exclusive option of Bellofram. In the event a claim should arise during the guarantee period, Bellofram must be notified immediately and the defective product made available for field or factory inspection and disposition. Bellofram cannot and does not accept responsibility of any type for any of its products that have been subjected to improper installation, application, negligence, tampering or abuse. In any event, the liability of Bellofram for a defective product is limited to the cost of the product or a replacement therefor. This warranty is in lieu of any and all other warranties, express or implied. Bellofram Corporation does not warrant its products to be merchantable and does not warrant its products in any manner for any specific purpose or use, and Bellofram Corporation disclaims any liability for consequential damages of any nature.

Prices and specifications are subject to change without notice.

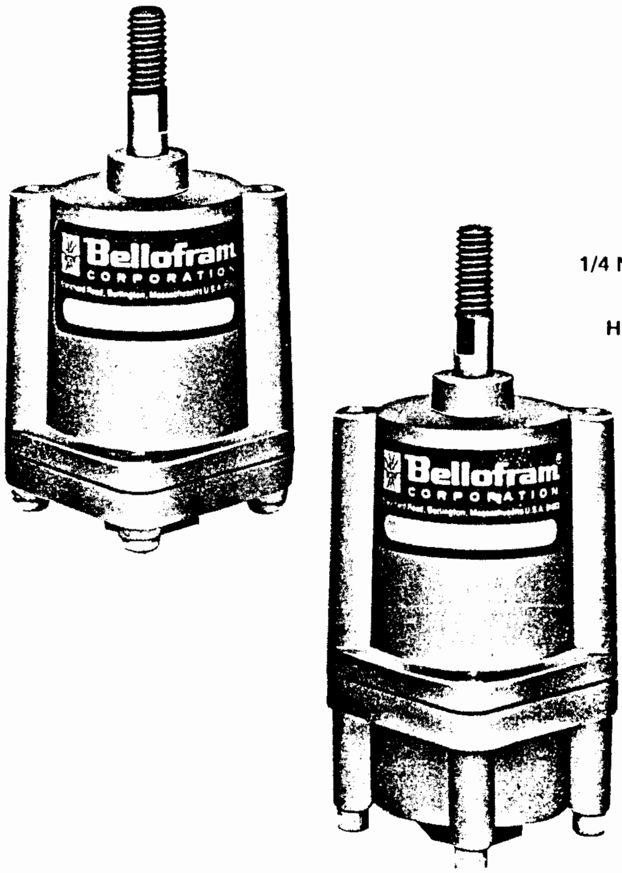
Low-Cost Cylinders

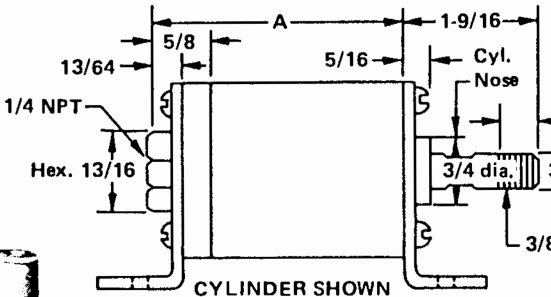
Foot Mounts

Order by P/N. 414-607-045

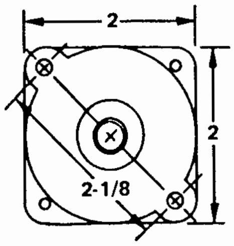


FOOT MOUNT BRACKET DIMENSIONS





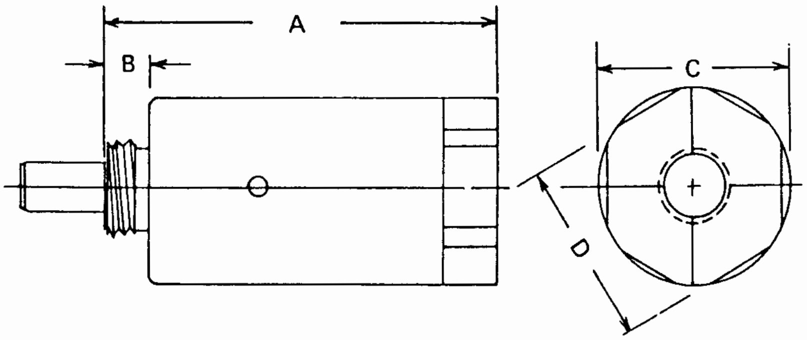
CYLINDER SHOWN WITH OPTIONAL FOOT MOUNTS

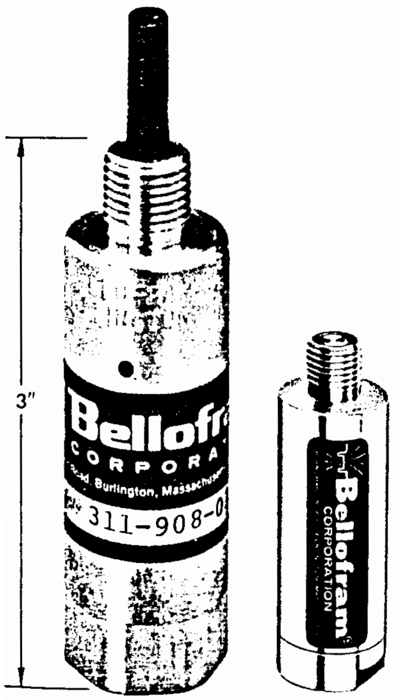


Specifications	Part Number	
	421-980-008	421-980-077
Stroke	1"	1-3/4"
Dimension A	2-29/32"	3-21/32"
Spring Load		
@ "O" Stroke	4 Lbs.	4 Lbs.
@ Max. Stroke	8 Lbs.	11 Lbs.
Equiv. Bore Dia.	1.5"	1.5"
Max. Oper. Pressure	125 PSI	125 PSI
Effective Area	1.7 in.^{2}	1.7 in.^{2}
Mounting Hole Sizes	1/4"-20 UNC	1/4"-20 UNC

Materials: Chrome-plated carbon steel rod. Die-cast aluminum cylinder and end cap. Oil-impregnated bronze bearing. Polyester fabric reinforced Nitrile diaphragm. Music wire spring.

Small Bore Cylinders





Now OEM's have the friction-free low hysteresis Air Cylinder they've dreamed of. Bellofram has applied its years of experience in manufacturing large sized friction-free Air Cylinders to provide OEM's with the first small bore diaphragm Air Cylinder available anywhere. A unique fabric reinforced rolling diaphragm guaranteeing friction-free seals has made the cylinder of your dreams come true. Available in selected standard sizes, with threaded or unthreaded rods, flush or extended; single-acting only, with spring return.

SPECIFICATIONS	PART NUMBER			
	311-908-013	311-908-034	311-908-014	311-908-035
A	2.81"	2.81"	1.95"	1.95"
B	0.438"	0.438"	0.244"	0.244"
C	15/16"	15/16"	15/16"	15/16"
D	7/8"	7/8"	7/8"	7/8"
Stroke	0.70"	0.70"	0.32"	0.32"
Spring Load				
@ "O" Stroke	2 lbs.	2 lbs.	5 lbs.	5 lbs.
@ Max. Stroke	7 lbs.	7 lbs.	7 lbs.	7 lbs.
Equiv. Bore Dia.	0.7"	0.7"	0.7"	0.7"
Max. Op. Pressure	125 PSI	125 PSI	125 PSI	125 PSI
Eff. Pressure Area	0.384 in.^{2}	0.384 in.^{2}	0.384 in.^{2}	0.384 in.^{2}
Nose Mount				
Thread Size	1/2"-20 UNF	1/2"-20 UNF	1/2"-20 UNF	1/2"-20 UNF
Rod Dia.	1/4"	1/4"	1/4"	1/4"
Rod				
Extension	Flush	3/4"	Flush	3/4"
Thread	–	1/4"-28 UNF	–	1/4"-28 UNF
Pipe Conn.	1/8"-27 NPSF	1/8"-27 NPSF	1/8"-27 NPSF	1/8"-27 NPSF

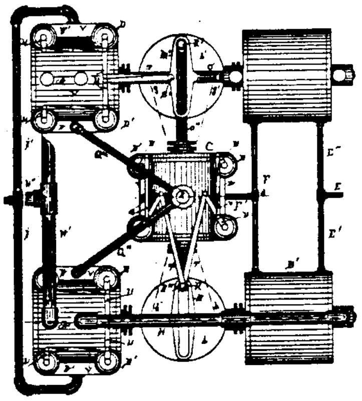
Materials: Carbon steel rod, aluminum alloy cylinder and end caps. Polyester fabric reinforced Nitrile diaphragm. Music wire spring. Oil-impregnated bronze bearing. Negligible breakaway friction.

U. S. PATENT OFFICE.

OFFICIAL GAZETTE.

AUGUST 30, 1904.

768,691. AIR-ENGINE. WILSON R. PRATT, Topeka, Kans. Filed Dec. 28, 1901. Serial No. 87,603. (No model.)



Claim.—1. The combination of straight-line engines, jackets provided on said straight-line engines, air-compressors communicating with said jackets, said compressors delivering to said straight-line engines, jackets provided on said air-compressors, an oscillating primary engine adapted to exhaust into said straight-line engines, and receiving air from said last-named jackets, substantially as described.

2. The combination with an oscillating primary engine, and its piston-rod, of rotating crank-disks, crank-pins on said disks, said piston-rod being connected to the said pins, double-bow stirrups in which the said crank-pins are arranged, straight-line engines, studs arranged in said bow-stirrups and connected with the said straight-line engines, and an air-compressor connected with said stirrups, substantially as described.

3. The combination with a high-pressure oscillating air-engine, of low-pressure straight-line air-engines receiving the exhaust therefrom, compressors driven by said engines, jackets for the compressors and straight-line engines, means whereby the compressors deliver to the straight-line-engine jackets, and means for conducting the air for the oscillating engine through the compressor-jackets, substantially as described.

4. In a device of the character described, the combination of an oscillating engine, straight-line engines adapted to receive the exhaust therefrom, means for compressing air adapted to be actuated by said engines, said means provided with air-jackets, and means for conducting the air for said oscillating engine through the jacket of said compressing means, substantially as described.

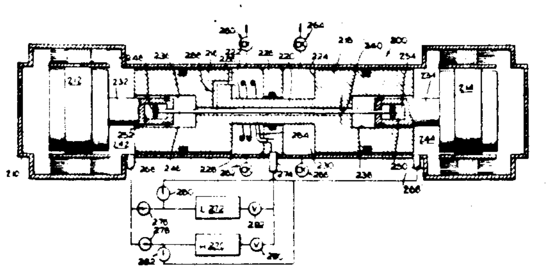
5. The combination with a primary high-pressure oscillating air-engine, of low-pressure straight-line engines receiving the exhaust therefrom, air-compressors actuated by said engines, and means whereby the compressed air by which the primary engine is actuated, is raised in temperature by the heat generated in the air-compressors, substantially as described.

6. The combination with a primary high-pressure oscillating air-engine, of low-pressure straight-line engines receiving the exhaust therefrom, air-compressors actuated by the said engines, semi-rotating valves arranged on the high and low pressure engines, means connected with said engines for operating said valves and means whereby the air by which the primary engine is actuated is caused to absorb heat generated in the compressors, substantially as described.

7. The combination with a high-pressure primary oscillating air-engine, of low-pressure straight-line engines receiving exhaust therefrom, jackets formed upon said last-named engines, compressors driven by said engines, jackets formed upon said compressors, means connecting said compressors with said jackets of the straight-line engines whereby the said compressors deliver thereto, means whereby the air for the oscillating engine passes through the said compressor-jackets, and means connecting said primary engine with the said straight-line engines and compressor-jackets.

JANUARY 5, 1971

3,552,120
STIRLING CYCLE TYPE THERMAL DEVICE
William T. Beale, Athens, Ohio, assignor to Research Corporation, New York, N.Y., a nonprofit corporation of New York
Feb. 1, 1967. This application Mar. 5, 1969, Ser. No. 812,530
Int. Cl. F03g *7/06;* **F25b** *9/00*
U.S. Cl. 60—24 **10 Claims**



A Stirling cycle thermal engine or refrigerating device wherein there is no primary mechanical connection between the displacer pistons and their associated power pistons, including various mechanical and pressure fluid means for varying the power output from or the power input to the device.

MACHINE DESIGN JUNE 21, 1984

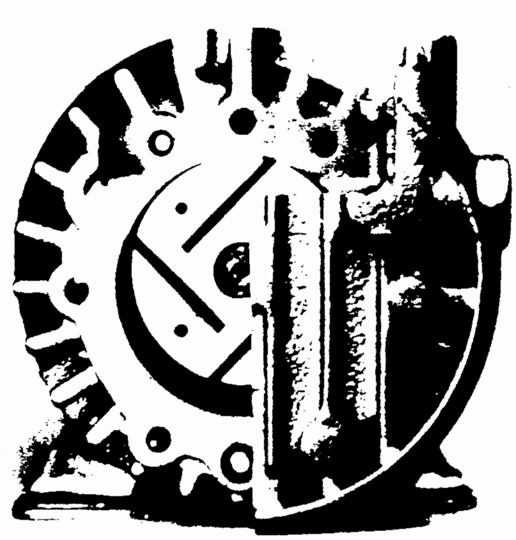
PNEUMATICS KICKS THE OIL HABIT

Once limited to low-pressure, low-flow applications, oil-less pneumatic systems are approaching the capabilities of lubricated systems.

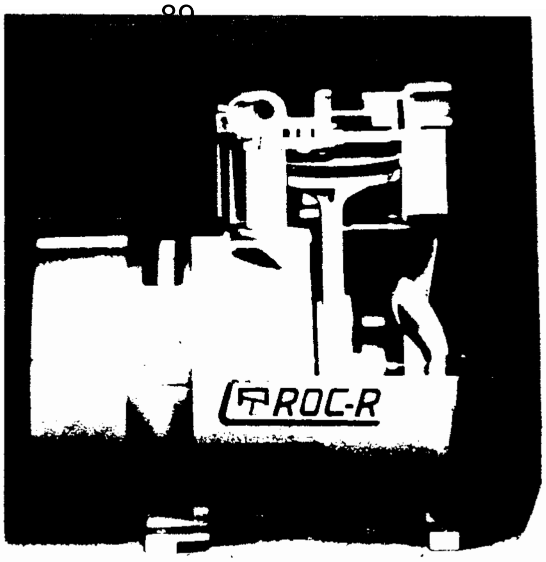
RICHARD C. BEERCHECK
Staff Editor

Oil is one of the biggest sources of contamination in a pneumatic system. It can damage the finished product, gum up components, and pose a hazard to personnel. To overcome these problems, manufacturers have developed a wide range of components that operate reliably without lubrication.

Three factors have spurred the move to oil-less pneumatic systems: the rising cost of maintenance and downtime, more stringent processing requirements for some products, and stricter government regulations on the amount of oil permitted in exhaust air.



Vane compressors were the first type converted to oil-less operation. The carbon vanes in this Gast Mfg. Corp. compressor are self-lubricating on cast iron in the presence of the water vapor commonly found in air.



Rocking piston compressors are a cross between reciprocating piston and diaphragm units. A self-lubricating PTFE cup on the piston of this Gast Mfg. Corp. compressor seals the compression chamber and reduces friction.

A typical plant compressed-air system can contain thousands of air-line lubricators. If each uses 8 oz of oil per week, the plant would need several people whose sole job is to fill the lubricators. No plant can afford this type of investment, and in actual practice most lubricators are serviced erratically. As a result, lubricators frequently run dry, leading to damaged valves, cylinders, and tools. To eliminate this cost and potential source of failure, many facilities are now demanding equipment that operates on oil-less air.

The second factor boosting the increased use of oil-less air systems is that many processes cannot tolerate even minute contamination of the finished product. Electronic-chip making, food processing, paper making, temperature controls, and textile manufacturing are typical processes that must avoid oil particulates in the exhaust air of pneumatic components.

Finally, OSHA now strictly limits the amount of oil mist permitted in the factory environment. And breathing air from medical equipment or safety suits must be free of oil to prevent injury or illness.

Oil-less compressors

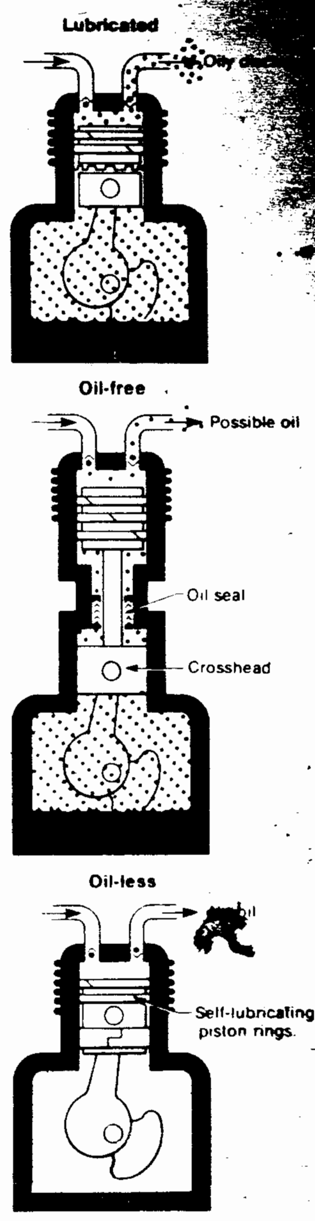
The heart of a compressed-air system is, of course, the compressor. Regardless of the type of compressor, the first design change required to make it oil-less is to use sealed bearings on the crankshaft and — for piston compressors — the wrist pin. Such bearings eliminate the need for splash or mist lubrication in the crankcase, the major source of oil particulates in a pneumatic system.

Getting the oil out

Reciprocating compressors are the most widely used source of compressed air for systems operating below 25 hp. Normally, compressors operating over about 5 hp must be lubricated to ensure long life and efficient operation. Lubricated compressors have oil-filled crankcases; thus, oil vapor is always present in the compression chamber and discharge line. Filtration to remove oil from the air line requires constant maintenance. Besides, no filters are 100% effective at removing oil.

The first attempt to eliminate oily discharge from reciprocating compressors were the so-called oil-free compressors. These units contain a crosshead and oil seals to isolate the oil-filled crankcase from the compression chamber. The best seals ensure 99.99% oil-free air, however, even these seals wear eventually, increasing the possibility of oil in the air.

Oil-less compressors, as their name implies, use no oil at all. They feature self-lubricating piston rings and skirts, as well as sealed bearings. With no oil in the crankcase, none can be entrained in the discharge air. Problems with heat generation presently limit the horsepower capacity of oil-less compressors to about 15 hp.



The main problems to overcome in switching from lubricated to oil-less operation are friction, heat, and higher noise levels. Friction has been reduced effectively through the use of self-lubricating materials. However, the temperature problem has limited the capabilities of oil-less compressors to about 15 hp, although higher-power units are on the horizon. The power limitation results because present-day materials cannot withstand the surface speeds and heat of higher-power operation. Noise problems are reduced by mounting the compressors in nonsensitive areas or by using readily available sound attenuating materials.

Besides power, the other major limitations of oil-less compressors is life. Presently, the sealed, grease-lubricated crankshaft bearings are rated for 8,000 to 12,000 h. Used in a typical 50% duty-cycle installation, these bearings provide about three years of operation before service is necessary. Similarly, the PTFE seals typically used in oil-less compressors are rated for 8,000 to 10,000 h of life.

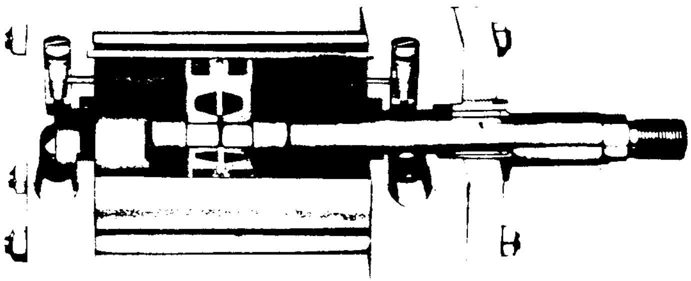
One of the first oil-less compressors was a vane compressor in which the phenolic vanes were replaced by carbon vanes. With the humidity commonly present in atmospheric air, carbon vanes are self-lubricating on cast iron. Over the years, the composition of the vanes has been improved to increase life; however, oil-less vane compressors are limited to a maximum operating pressure of about 15 psi and a maximum power of about 5 hp.

Diaphragm compressors, of course, have always provided oil-less operation because the diaphragm completely isolates the crankcase from the compression chamber. These compressors can produce up to 100 psi; however, their limited stroke restricts them to low-flow, low-power installations.

Oil-less piston compressors were developed in the mid-1950s. At first, these compressors used split carbon rings to seal the cylinder bore. However, such rings wore after only a few hours of operation, widening the ring gap and reducing flow capacity.

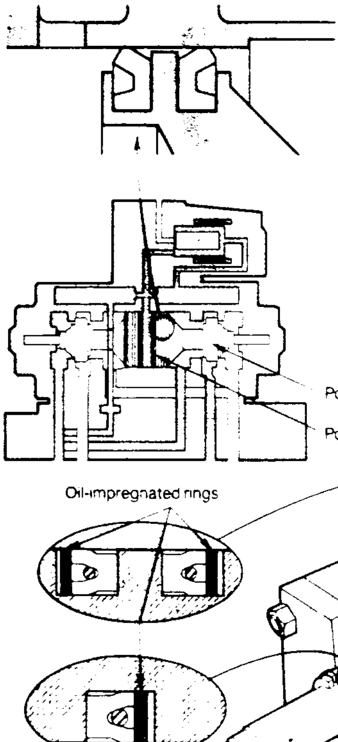
The problem of wearing rings has been overcome in most cases through the use of PTFE piston rings and skirts. PTFE rings effectively seal the cylinder bore and reduce friction. However, while such compressors can provide up to 100 psi, power is still limited.

The factor limiting the capacity of piston compressors is the

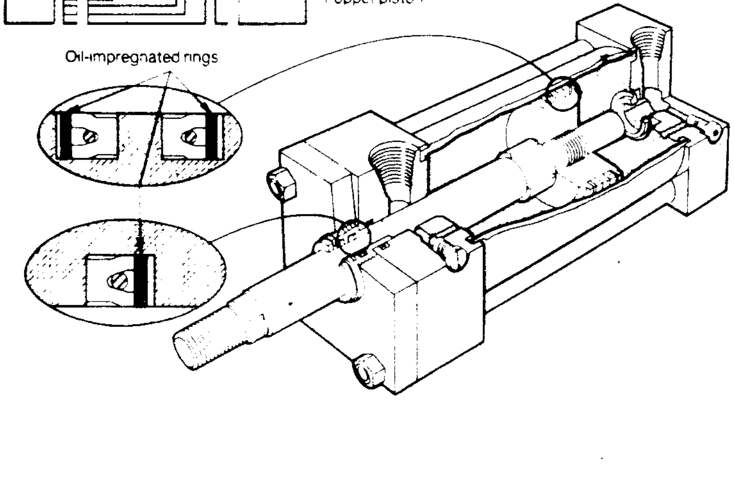


Oil reservoir stores lubricant in the piston of this cylinder from Lehigh Fluid Power Inc. The oil is wicked to the cylinder wall and is retained there by lip seals. An oil-impregnated bushing lubricates the piston rod.

Friction and heat are the biggest problems limiting the capacity of oil-less compressors. This compressor from ITT Pneumotive reduces friction and heat generation with force-compensated PTFE piston rings and skirts. An integral blower and shroud (not shown) direct cooling air over the finned cylinders and crankcase.



Poppet valves are widely used in oil-less air systems because they have minimal sliding contact. Such valves typically are prelubricated at assembly. Special lip seals, such as those on this valve from Parker Hannifin Corp., retain the lubricant in the poppet piston area, preventing it from being exhausted downstream. Some valves include a lubricant reservoir between the lip seals.



Self-lubricating cylinder from Parker Hannifin Corp. has oil-impregnated rings in the piston and rod gland. The rings dispense oil to the cylinder wall and rod. Lip seals retain the oil in these areas and prevent it from being exhausted to the atmosphere.

heat produced in the compression process, which reduces the life of the bearings and seals. To boost capacity, manufacturers of piston compressors have devised innovative ways to reduce heat buildup and remove the heat that does develop.

The usual way to reduce heat generation is with so-called force-compensated piston rings. The rings ride on a cushion of air, sealing during the compression stroke but releasing on the intake stroke. This mode of operation reduces the friction and pressure forces on the rings, cutting heat generation and improving wear life. Any heat that is generated is removed by forced-air cooling. A blower wheel mounted on the end of the motor shaft directs cooling air through the crankcase and over the pistons, cylinders, and bearings, permitting the compressors to operate continuously at up to 15 hp and pressures to 200 psi.

A third type of oil-less reciprocating compressor, the rocking-piston type, is a cross between piston and diaphragm compressor. The piston is sealed by a PTFE cup that expands as it is forced into the cylinder. The expansion compensates for the rocking motion of the piston and maintains a tight seal on the cylinder wall. Rocking-piston compressors can produce up to 100 psi, but maximum power rating is under 1 hp.

Nonlubricated valves

The designation "nonlubricated" for an air valve usually means that the valve is prelubricated or contains parts made of a self-lubricating material. Valves are prelubricated in one of two ways: 1. The valve parts are dipped in a lubricant prior to assembly. 2. The valve includes a reservoir of lubricant that is fed to critical areas. In either case, the valves normally do not require further lubrication in service.

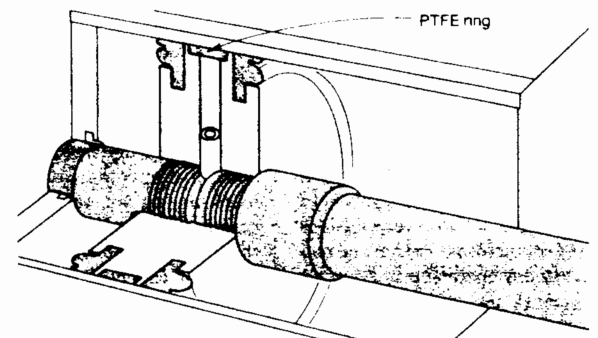
The design of pneumatic directional-control valves to operate without air-line lubrication requires special considerations. And there is considerable debate over the types of valves suitable for oil-less service. For instance, some experts contend that spool valves, with sliding action between parts, can experience problems with wear and high shifting forces when the lubricant either wears off or is carried away by the air stream. Also, past experience with packed-spool valves has shown that the synthetic rubber seals commonly used in such valves can actually wear away the metallic sealing surfaces if the valves are not lubricated.

However, some manufacturers of packed-spool valves replace the rubber seals with self-lubricating PTFE seals, and they hardcoat the valve bodies. This combination provides a low-friction, high-lubricity sliding contact that supporters claim provides reliable operation on nonlubricated air.

In general, lapped-spool and poppet valves are the most widely used types for oil-less service. Lapped-spool valves depend on the close fit between body and spool to seal the valve. The valves typically are dipped in a lubricant prior to assembly and require no further lubrication in service.

However, lapped-spool valves can have problems when operating on nonlubricated air, because they are susceptible to clogging from contaminants or from hydrocarbons carried downstream from a lubricated compressor. Hydrocarbons are particularly troublesome, because the heat of compression transforms them into varnishes that can build up in the valve. When the valve sits idle for long periods, such as over a weekend, the varnish can cause the spool to bind, preventing proper shifting and resulting in solenoid burnout. Manufacturers claim that lapped-spool valves are easily protected with coalescing filters. Such filters effectively remove hydrocarbon aerosols and solid particles from the air.

In contrast to other valve



PTFE ring on the piston of this Mosier Industries Inc. cylinder lubricates the piston/cylinder-wall contact. The cylinder wall is highly polished to reduce friction, and an oil-impregnated bushing lubricates the piston rod.

types, poppet valves have no sliding action in the primary sealing area. Thus, the valves resist clogging from air-line varnish and contaminants. The only area of a poppet valve with sliding contact is the piston or actuation section. But this area can be prelubricated to operate for millions of cycles through the use of either self-compensating seals or an internal lubricant pocket.

Valves designed for normal service usually are prelubricated and have self-compensating seals. In such valves, lip seals on the piston are specially designed to hold the lubrication in place, and they wipe the bore clean with each stroke. Because the seal lips are self-compensating, sealing is maintained even as components wear during operation. Such valves also can have bodies made of self-lubricating injection-molded plastic, further reducing the need for lubrication.

Valves for heavier-duty service usually include a lubricant reservoir between the piston lip seals. The lubricant is fed continuously to the piston/cylinder-wall interface and is retained in this area by the lip seals. In tests, such valves have survived over 100 million cycles on nonlubricated air.

Lubed-for-life cylinders

Over the years, there has been much misunderstanding about the true meaning of nonlubricated cylinder requirements. Basically, nonlubricated cylinders fall into three categories: prelubricated, unlubricated, and self-lubricating. Self-lubricating cylinders can be further categorized as those with lubricant reservoirs and those with self-lubricating parts.

Prelubricated cylinders are the oldest type of nonlubricated cylinder. At assembly, an extreme-pressure grease is applied to the seals, piston, cylinder bore, and gland surfaces. The lubricant adheres tightly to the cylinder parts, resisting explusion into the atmosphere. Tests of such cylinders have shown that the moisture commonly present in compressed air also helps to lubricate the piston seals.

Unlubricated cylinders are constructed of absolutely dry parts with no lubricant. Requirements for such cylinders arose in the food industry because of concerns about the possibility of contamination. However, unlubricated cylinders require special, expensive seals, and they have had only limited success in tests. Further evaluation of the possible use of unlubricated cylinders has shown that a truly unlubricated design is not absolutely necessary. As a result, the demand for this type of cylinder has never really developed.

Self-lubricating cylinders were developed to meet an auto-industry specification calling for cylinders to operate 20 million cycles on dry, moisture-free air. To survive such severe service, these cylinders incorporate either an internal lubricant reservoir or self-lubricating piston-ring and rod-seal materials.

There are several different approaches to providing a lubricant reservoir. In some cylinders, special composite rings impregnated with extreme-pressure oil are used in the piston and rod-seal areas. The rings are positioned to dispense oil to the cylinder bore and piston rod during each stroke. Special lip seals collect excess lubricant so that none escapes into the atmosphere.

Other cylinders incorporate an oil reservoir in the piston, which is filled at assembly. The lubricant is then wicked to the cylinder wall. Again, lip seals collect excess lubricant, preventing it from escaping to the atmosphere.

Cylinders with self-lubricating parts commonly include a phenolic or PTFE wear ring on the piston and either an oil-impregnated bushing or wear ring in the rod gland. The cylinder bore and rod also are highly polished to reduce friction. These cylinders can be either prelubricated or unlubricated at assembly.

Technical assistance provided by Gast Mfg. Corp., Benton Harbor, MI; Humphrey Products, Kalamazoo, MI; ITT Pneumotive, Monroe, LA; Parker Hannifin Corp., Pneumatic Div., Otsego, MI; Ross Operating Valve Co., Detroit, MI; Finite Filter Co. Inc., Oxford, MI; Lehigh Fluid Power Inc., Lambertville, NJ

Filtration — key to oil-less operation

Because of the power limitations of oil-less compressors, most large air systems continue to use lubricated compressors. Estimates of the hydrocarbon content in the discharge air of such compressors are:

- Screw — 25 to 75 ppm at 200°F.
- Reciprocating — 5 to 50 ppm at 350°F.
- Centrifugal — 5 to 15 ppm at 300°F.

At a concentration of only 25 ppm, a typical compressor flowing 100 scfm for only 35 h will dump 8 oz of oil into the pneumatic system. To prevent the oil from contaminating components or the finished product, a coalescing filter is essential.

Coalescing is a continuous air filtration process in which liquid aerosols contact glass microfibers and are agglomerated into larger and larger droplets. The droplets eventually emerge on the downstream side of the filter and are drained away.

The coalescing process occurs in three ways, depending on aerosol size and inertial force. Aerosols larger than 2 μm are collected by direct impact, with larger droplets being formed by successive impingements. Aerosols from 0.2 to 2 μm — the predominant size range — are collected by interception by the glass fibers. Aerosols in the 0.001 to 0.2 μm size range follow random Brownian motion, striking the fibers through diffusion and being held by van der Waals forces.

The standard coalesing filter removes 99.97% of the aerosols in the 0.3 to 0.6 μm range, is 99.98% efficient at removing all aerosols, and removes all solid particles larger than 0.3 μm. Thus, incoming air at a 20 ppm level is reduced to 0.004 ppm concentration, which is acceptable for almost any critical pneumatic system. More efficient filters rated at over 99.9999% aerosol removal also are available.

Coalescing filter is made of borosilicate glass microfibers vacuum formed into interlocking contact. This filter, from Finite Filter Co. Inc., has a rigid porous support tube and a glass matrix drain layer. It is compatible with all compressor lubricants, and removes oil aerosols and solid particles from the air stream.

TORQUE CONVERTERS / CHAPTER 6

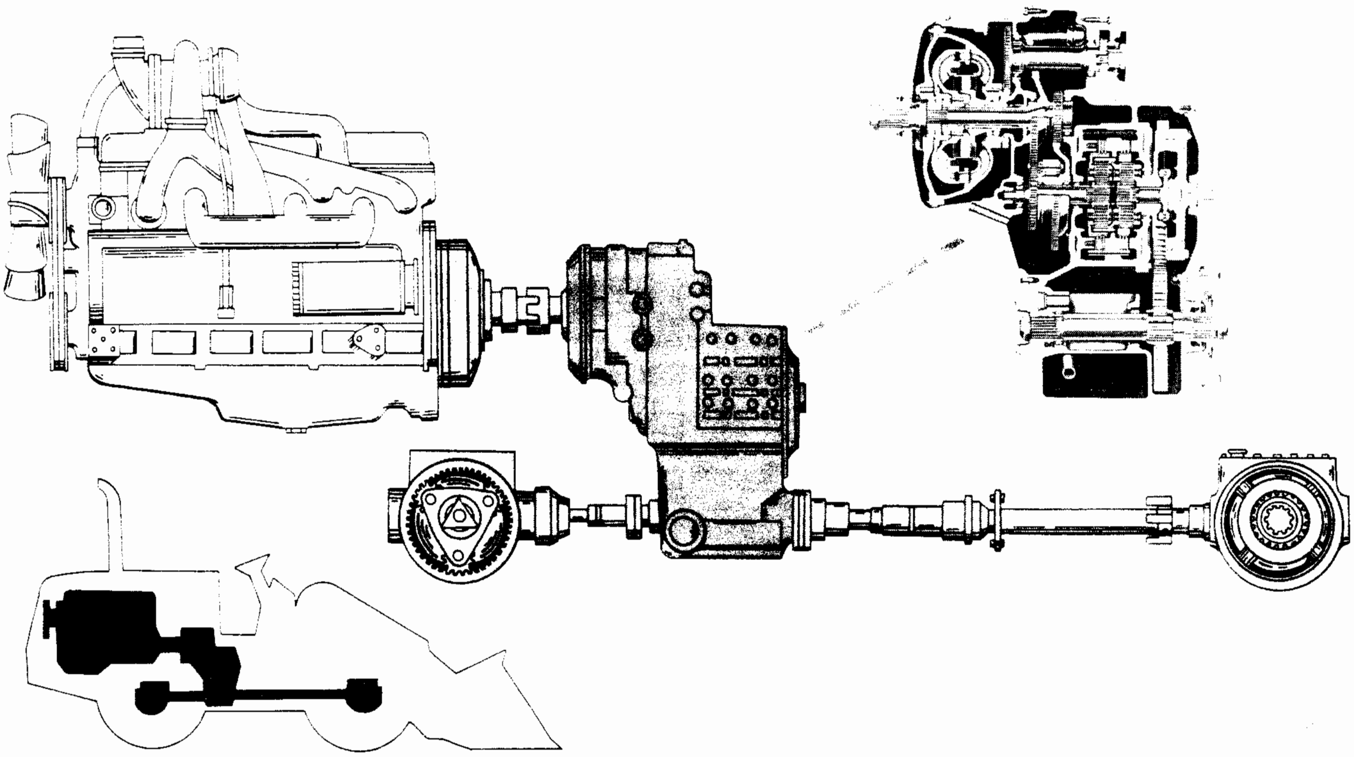


Fig. 1—Torque Converter In Complete Power Train

INTRODUCTION

A **torque converter** is an automatic *fluid* drive. It transmits engine torque by means of hydraulic force, shifting smoothly through an infinite number of speeds.

The automatic transmission of an automobile automatically shifts gears in response to torque requirements in addition to the automatic response of the torque converter which is a part of the automobile's automatic transmission system.

Actually a gear train is used with the torque converter to give extra speed ranges. But no gear train could give the infinite variations in speed and torque of a torque converter.

Acting as a *clutch*, the torque converter connects and disconnects power between the engine and the gear train. As a *transmission*, the converter gives many more speed ratios than are practical with a strictly mechanical gear box.

To compare a torque converter with a hydrostatic drive (Chapter 5), use this rule of thumb:

Hydrostatic drives are driven by fluids at *high pressure* but at relatively *low velocity*, while **torque converters** are driven by fluids at *low pressure* but at *high velocity*.

Here are the formulas:

- **Hydrostatic Drive** = *HIGH pressure* - *LOW velocity*
- **Torque Converter** = *LOW pressure* - *HIGH velocity*

HOW IT WORKS

To understand a torque converter, we must first look at a basic **fluid coupling.**

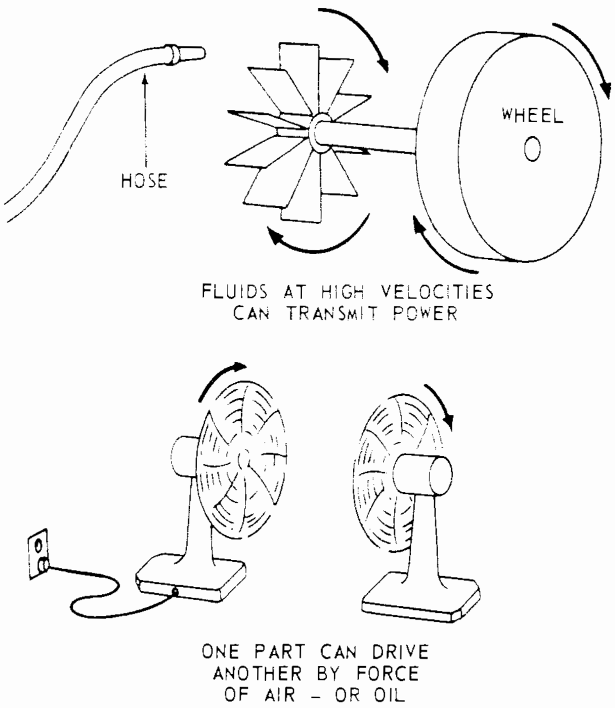


Fig. 2—Basic Principles Of A Fluid Coupling

The basic principles of all fluid couplings are shown in Fig. 2.

At the top, a fluid at high velocity strikes a turbine and forces it to turn, driving the wheel. Thus, **torque is transmitted by a fluid.**

To change this torque, the velocity of the fluid is changed. At low velocity, the fluid will not even move the turbine. At higher velocity, the turbine starts turning and the wheel picks up speed.

This is something like putting two electric fans face to face as shown in the lower part of Fig. 2. By plugging in only one fan, we can cause the other one to rotate.

This principle is used in a fluid coupling as follows:

Inside an oil-filled housing (A, (Fig. 3) are two parts: the driving half, or **pump** (impeller), and the driven half, or **turbine**.

As the pump is turned by the engine, centrifugal force causes oil to be forced radially outward, crossing over and striking the vanes of the turbine. This rotates the turbine in the same direction and so couples the power.

Drawings B, C, and D explain how the flow of oil drives the turbine.

In B, fluid is placed in a bowl and lies level.

In C, the bowl is spun rapidly and centrifugal force causes the fluid to climb up and spill over the outside edge of the bowl.

In D, another bowl is placed down over the first one. Now when the bowls are spun an axial flow or circuit is created and turning force is transmitted between the driving bowl and the driven bowl.

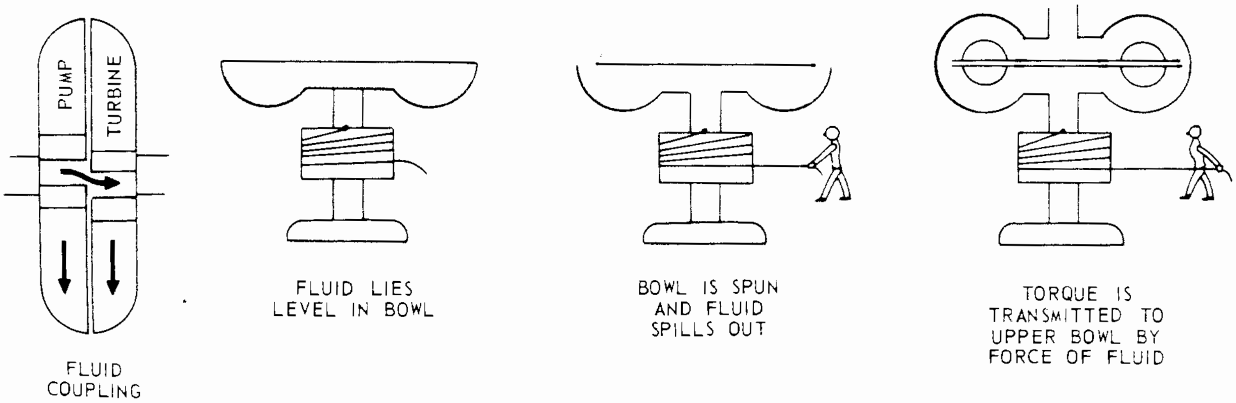


Fig. 3—Operation Of A Fluid Coupling

Litho in U.S.A.

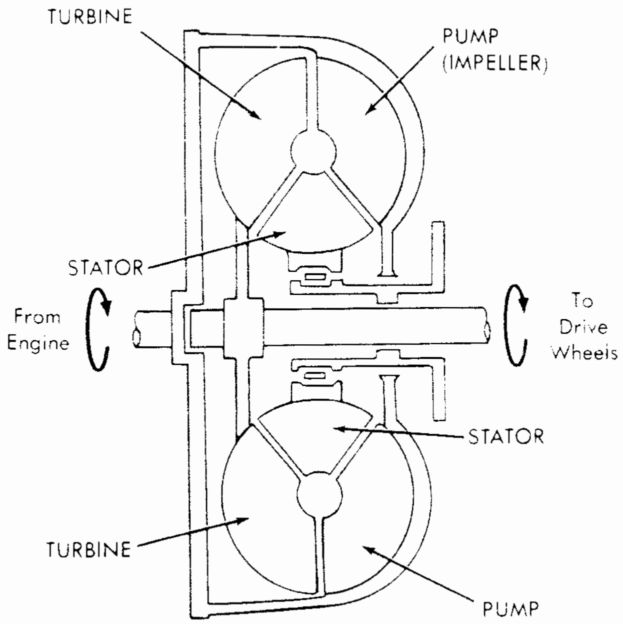


Fig. 4—Torque Converter

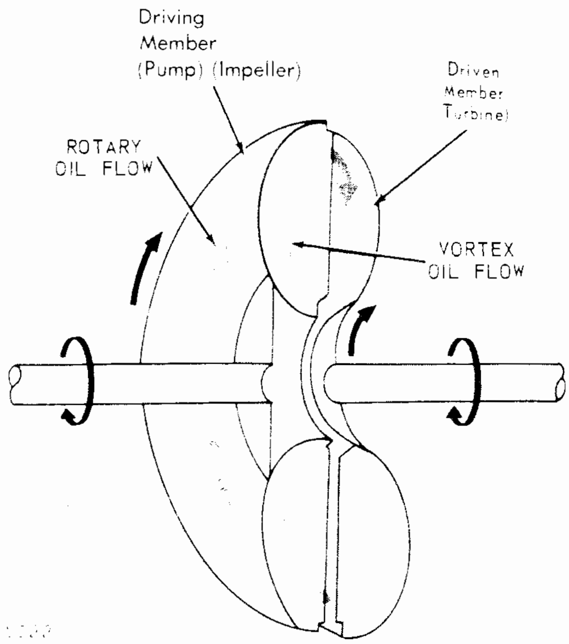


Fig. 5—Vortex And Rotary Flow In A Fluid Coupling

Torque is thus transmitted, but it is not increased.

This is where the **torque converter** goes beyond the basic fluid coupling, for the converter can **multiply torque.**

A torque converter (Fig. 4) looks much like the fluid coupling we have just described. The main difference is that the torque converter has—in addition to the driving pump and the driven turbine —a set of blades or vanes called a **stator.**

The stator vanes change the direction of oil flow after it has gone through the turbine and sends it back to the pump. This enables the pump to increase the twisting force or *multiply* the torque.

Since the converter is a closed unit, this flow is repeated continuously. Many streams of fluid act against many vanes at once and this is what gives the power to drive a heavy machine.

OIL FLOW IN CONVERTER

Let's look at the flow of oil in the converter during two cycles:

1) Increasing the torque

2) Decreasing the torque

INCREASING THE TORQUE

Remember that the pump is driven by the engine, while the turbine receives fluid energy from the pump and sends it to the drive wheels.

Also remember how centrifugal force sets up a continuous circular flow in the coupling (Fig. 5).

This circular flow of oil *between* the pump and turbine is called **vortex flow.**

Another flow is set up *around* the pump and turbine to form a coupling: this is called a **rotary flow.**

The action of these combined oil flows will transmit torque *but not increase it.*

Increasing the torque is where the stator comes into play.

Litho in U.S.A.

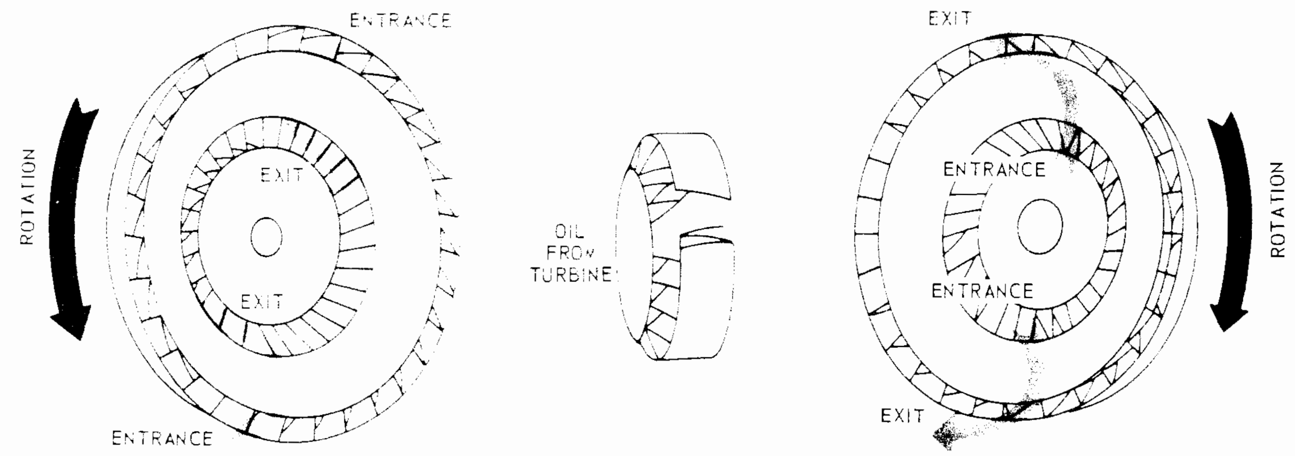


Fig. 6—Oil Flow Through Pump, Turbine, And Stator

Fig. 6 shows how oil emerges from the turbine in reverse compared to pump rotation. Unless this oil flow is turned around, it will cause a loss of power.

Note in Fig. 6 that the oil passages at the rim of the turbine where the oil enters must become smaller as they approach the smaller diameter of the turbine. As the same volume of oil must squeeze through these funnel-like passages, the oil stream will speed up when it leaves the turbine. This speed is used to increase torque by directing it against the **stator**, which acts as a fluid lever or fulcrum. The stator changes the direction of flow and sends the oil into the pump in the same direction as pump rotation.

Let's see how the stator does its job (Fig. 7). A stream of oil aimed at a flat surface (A) splashes off at various angles. The oil can be made to flow more smoothly by curving the entrance (B) and can be reversed by more curving (C) with a resulting increase in force as indicated by the large arrow.

The stator has curved blades (as in C) which the oil strikes as it leaves the turbine. These blades turn the oil back in the direction of pump rotation.

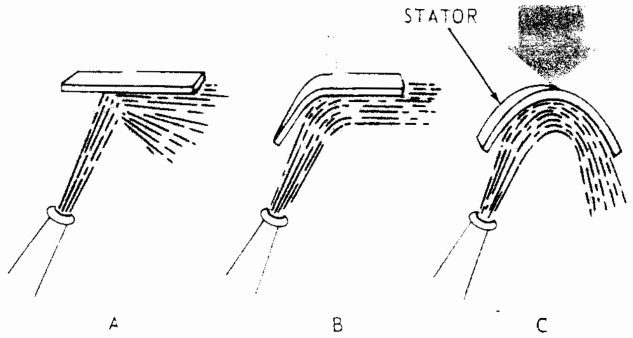


Fig. 7—Stator As A Fluid Lever

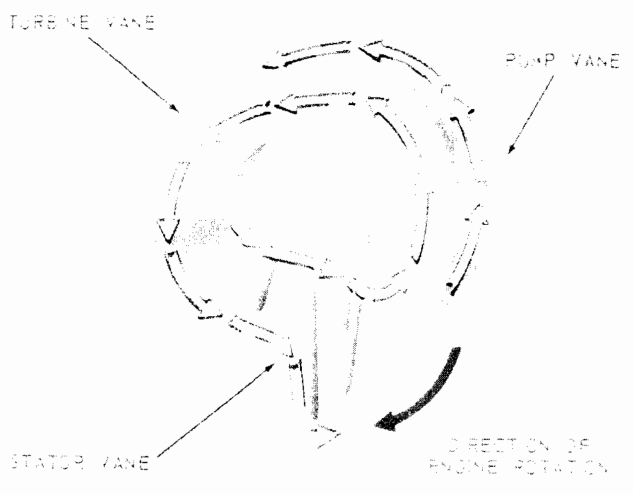


Fig. 8—Flow Direction Is Reversed By The Stator Vanes

Now that the stream of oil is moving in the same direction but at a greater velocity, it enters the pump smoothly (Fig. 8). Its velocity is added to that developed in the pump so that the total velocity at the pump exits has been increased.

This regenerating action is the key to multiplying torque in a torque converter.

To change the direction of oil flow, the stator must be stationary during the increase in torque. However, once the pump and turbine are turning at the same speed it would create resistance. Therefore, the stator is sometimes mounted on a freewheel clutch so that it can turn in one direction only (once torque stops increasing). (In other torque converters, the stator may be fixed to the converter case.)

DECREASING THE TORQUE

Torque is increased as long as the engine is accelerating to get the machine under way. But as the engine speeds up, the turbine also speeds up, which causes the vortex flow of oil in the converter to decrease. At the same time, the rotary flow increases.

Vortex flow keeps on changing to *rotary* flow (Fig. 9) until the pump and turbine are "locked up" and all torque increase stops.

The torque converter now acts as a simple fluid coupling, sending the same torque it receives on to the drive wheels.

The torque converter is able to automatically reduce or increase torque in infinitely small steps to match the needs of the machine and its driver. This has the same effect as shifting speeds in a gear transmission except that it is done smoothly and automatically while "on the go."

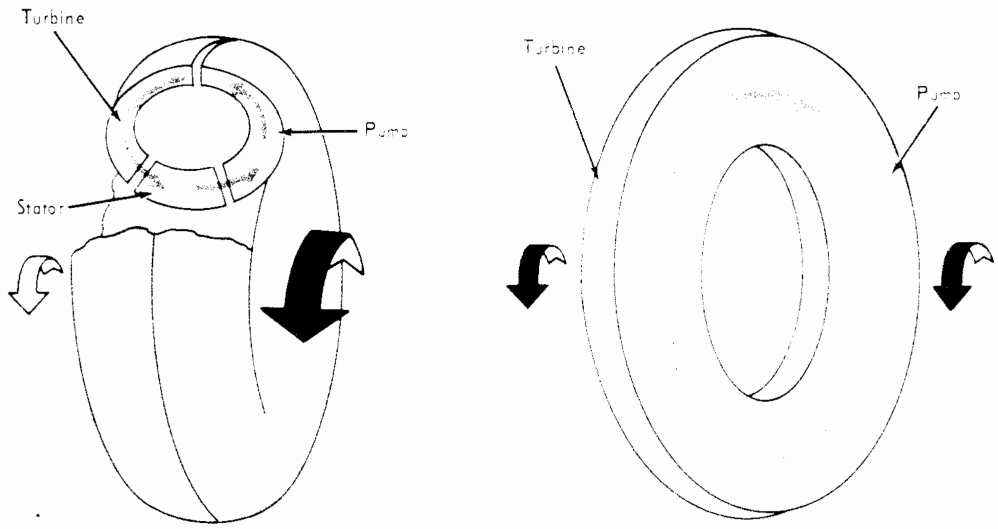


Fig. 9—How Vortex Flow Changes To Rotary Flow As Torque Is Reduced

VARIATIONS ON TORQUE CONVERTERS

The chart below shows some of the variation in the number of elements used in several torque converters.

Torque Converter Elements	*Design A*	*Design B*	*Design C*	*Design D*
Pump	*2*	*1*	*1*	*1*
Stator	*2*	*1*	*2*	*1*
Turbine	*1*	*2*	*1*	*1*

The design of a torque converter must match the engine torque and road speed for each application. In off-the-road machinery, units are matched to horsepowers (kilowatts) ranging from 40 to 600 (30-450). But the basic principles we've covered apply to all.

TORQUE CONVERTER TRANSMISSIONS

The torque converter is but one part of the complete transmission (Fig. 10).

Here are the major components:

- **Converter Section**
- **Range Gear Section**
- **Final Drive Section**
- **Hydraulic Control System**

Let's look at each part and see what is does for the complete transmission.

CONVERTER SECTION

We have looked at torque converters with three elements—one pump, one turbine, and one stator.

Now we will examine a *twin-turbine* model (Fig. 11) which has one pump and one stator, but *two* turbines (first and second).

The first turbine is shown in blue, while the second is shown in red.

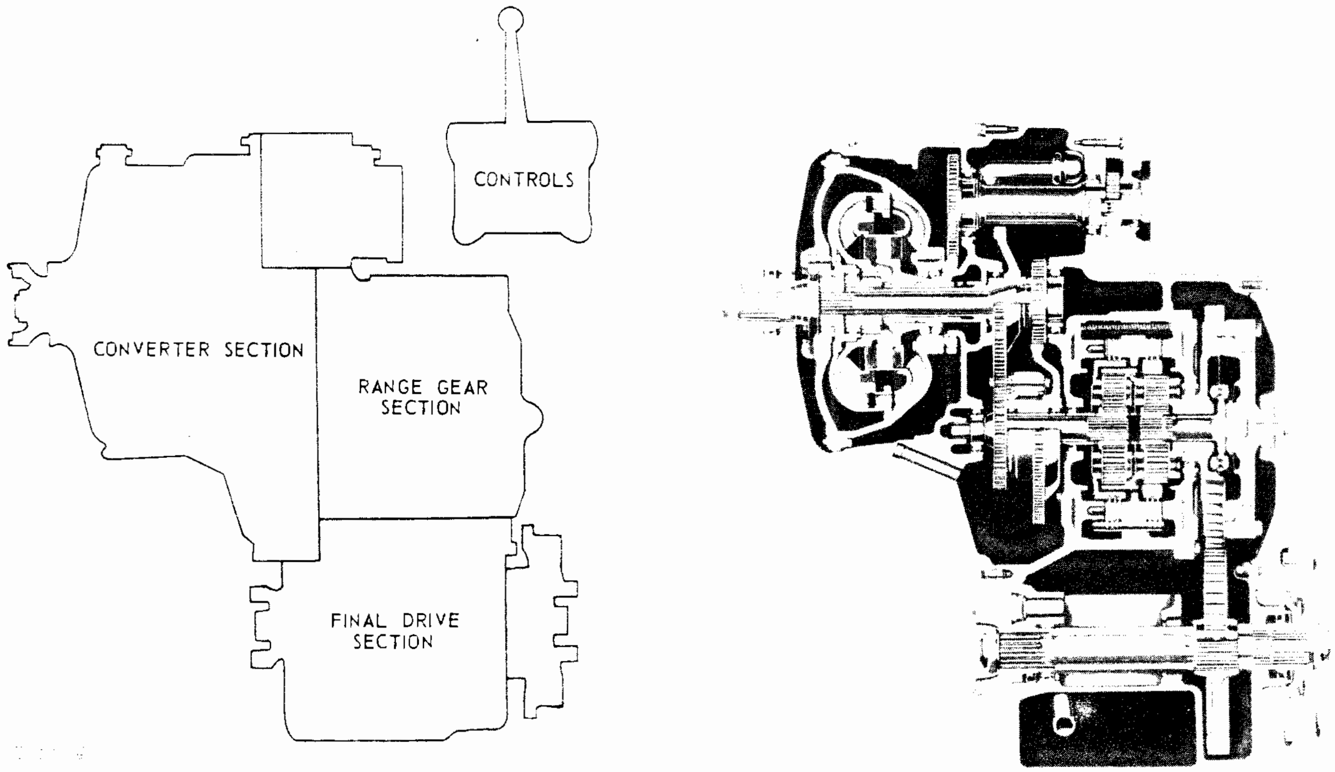


Fig. 10—Complete Torque Converter Transmission

Each turbine is connected to its individual output gear set. In reality, two converters have been combined into one. The first (blue) turbine is connected to its output shaft by a freewheel clutch.

Here is how the two turbines work together:

When torque demand is high, the freewheel clutch is engaged and the first turbine, assisted by the second turbine, drives the gears. When the machine speeds up, torque demand drops. Then the second turbine takes over the entire load and the freewheel clutch disengages the first turbine.

As a result, the first turbine provides *high torque and low speed*, (for starting up and loading) while the second turbine provides *higher speed with lower torque* (for travel).

A combining gear set directs torque from the first and second turbine (or the second turbine only) to the range gear section.

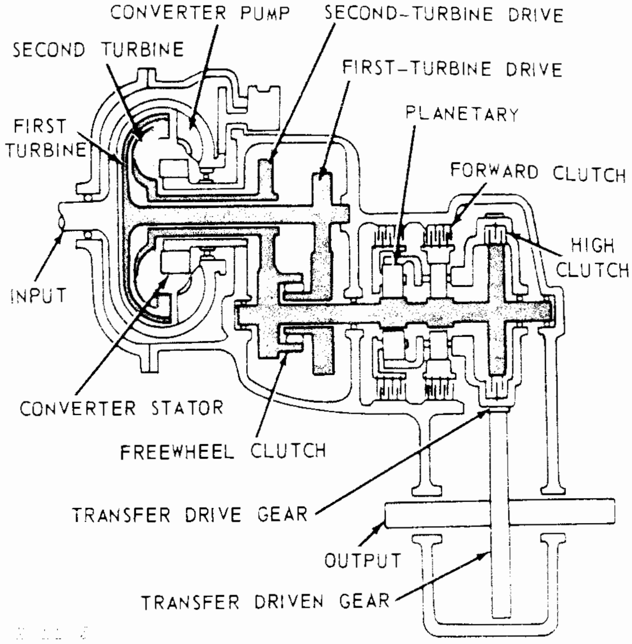


Fig. 11—Twin-Turbine Torque Converter

RANGE GEAR SECTION

Since torque is reduced and increased automatically in the torque converter, only a few gear sets are normally required in the transmission.

However, as the torque converter rotates in only one direction, it is necessary to have a reverse gear. In some applications, it is also desirable to have low and high range gears as shown in Fig. 11.

Simple planetary gear sets meet the needs of extra gear ranges and lends themselves to hydraulic control.

NOTE: Planetary gear sets and their control by hydraulic clutches are explained in Chapter 4.

FINAL DRIVE SECTION

The final drive section includes the transfer drive gear, transfer driven gear and output shaft (Fig. 11). The output shaft provides for one or for two outputs from the same common shaft. Two outputs can be used to propel a four-wheel drive machine as shown in Figs. 11 and 15.

Note also that by adding the transfer gear and one clutch (high range) in Fig. 11, another forward speed range can be obtained.

HYDRAULIC CONTROL SYSTEM

The hydraulic control system uses oil to do these jobs:

- **Oil flow lubricates and cools the parts**
- **Oil pressure engages the clutches**
- **Oil velocity drives the turbines**

Let's use the hydraulic controls for the twin-turbine converter we have just described to see how a typical system works.

There are four basic circuits as shown in Fig. 12:

1) Oil Pump and Filter Circuit (shown by blue lines).

2) Main-Pressure Regulator Valve and Converter-In Circuit (shown by red lines).

3) Converter-Out, Cooler, and Lubrication Circuit (shown by dotted blue lines).

4) Selector Control Valve Circuit (shown by dotted red lines).

Let's build up the system and explain each circuit.

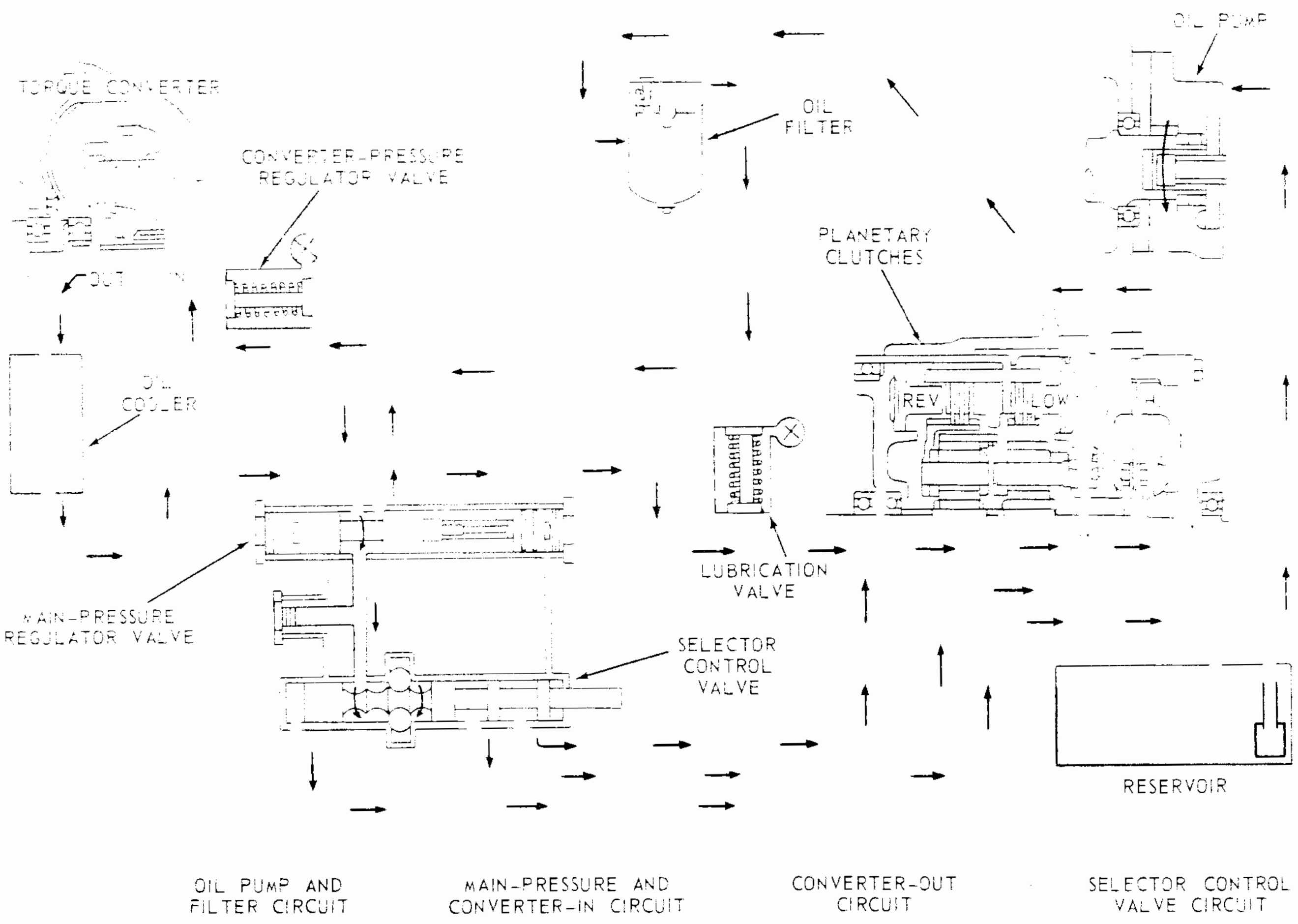


Fig. 12—Hydraulic Control System For Torque Converter Transmission

Oil Pump And Filter Circuit

Oil is drawn from the transmission reservoir by the oil pump as shown in Fig. 12. The pump delivers its entire output to a full-flow oil filter for cleaning. From the oil filter, the oil supply is sent to the main-pressure circuit.

Main Pressure Regulator Valve and Converter-In Circuit

The main pressure regulator valve provides pressure for the planetary clutch packs, directs oil to the selector control valve, and supplies pressure oil into the torque converter. A converter pressure regulator valve in the converter-in line limits the oil pressure there.

Converter-Out, Cooler, and Lubrication Circuit

The torque converter is continuously filled with oil during operation. Rotation of the converter pump imparts energy to the oil which, in turn, drives the turbines. The oil then flows between the stator vanes which redirect it to the pump.

Oil flowing out of the converter is directed into the oil cooler as shown in Fig. 12. The cooler is a heat exchanger in which the oil flows through water- or air-cooled passages.

From the cooler, oil flows to all passages and outlets in the lubrication circuit (dotted blue lines). A lubrication valve between the cooler and lubrication system returns all excess oil to the transmission reservoir.

Selector Valve Circuit

Pressure oil from the main-pressure regulator valve flows into the selector valve bore and surrounds the valve in the area of the detent notches. From this area, main pressure oil is available for operating the low, high, and reverse range planetary clutches (see dotted red lines).

Moving the selector valve allows oil to charge the selected clutch line and to engage that clutch.

This completes the four basic control circuits in our torque converter transmission.

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NOTE: For details on hydraulic components, see the FOS "Hydraulics" manual.

SUMMARY: FEATURES OF TORQUE CONVERTERS

1. Multiply torque

2. Provide infinite speed ranges

3. Shift smoothly and automatically

4. Cushion shock loads on drive lines

5. Help to dampen vibrations

TROUBLESHOOTING

INTRODUCTION

Oil circulates at high velocity within the torque converter and any foreign material it carries will rapidly wear down the edges and pit the turbine vanes (Fig. 13), changing their effective shape.

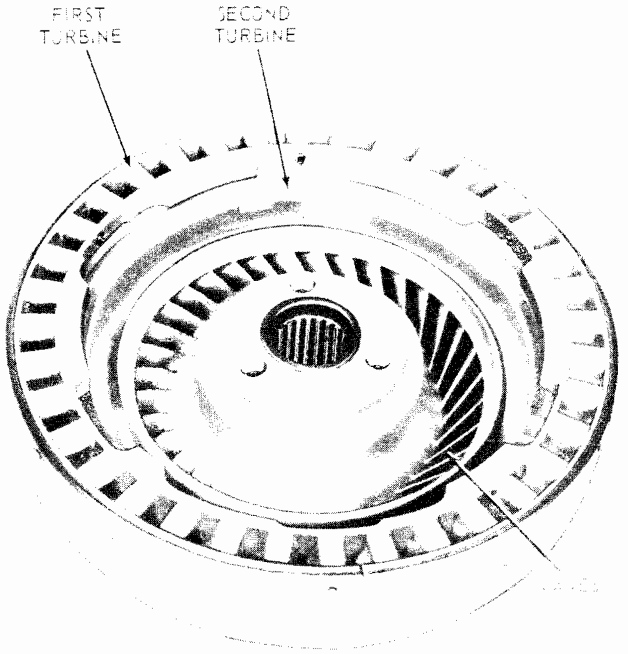

Fig. 13—Turbines For Twin-Turbine Torque Converter

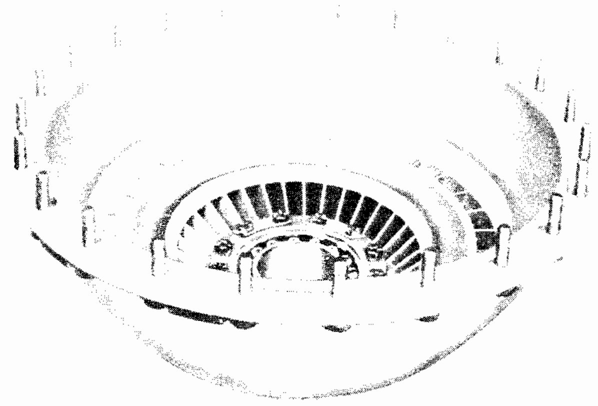

Fig. 14—Pump For Torque Converter

Vane damage will also cause the turbines to become unbalanced. In addition, dirty oil will damage bearings and seals.

Some torque converters contain parts made of lightweight aluminum alloys (Fig. 14). Converter housings are usually made of cast aluminum.

Be sure to handle all converter parts carefully to prevent nicking, scratching, and denting.

Parts which fit together closely but with operating clearance will stick if damaged only slightly. Parts which depend upon smooth surfaces for sealing may leak if scratched.

All these parts should be carefully handled and protected during removal, cleaning, inspection, and installation.

Use these rules to help prevent failures:

1. Be sure the oil is kept clean.

2. Service the system at proper intervals.

3. Repair it only if you are qualified.

4. Use all special tools recommended.

5. See the machine Technical Manual for details.