IZMIR INSTITUTE OF TECHNOLOGY

Department of Mechanical Engineering

ME311 Machine Elements 1

Chain Design Report

HW3

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1. Introduction

Chain drive systems are widely used in industrial applications where reliable power transmission at relatively low speeds is required. Compared to belt drives, chain drives provide a positive transmission without slip, making them suitable for systems in which speed accuracy and load consistency are critical. However, the correct selection of chain size, sprocket geometry, and lubrication method is essential to ensure long service life and safe operation.

In this study, a chain transmission system is designed based on parameters derived from the student identification number. The design process follows standard catalogue-based procedures, where power capacity, speed ratio, service conditions, and geometric constraints are evaluated step by step. Rather than selecting components directly from tables, the design is guided by limiting conditions such as allowable driven speed range, service time, and center distance.

The primary objective of this assignment is to transmit the required power from the driving shaft to the driven shaft while maintaining the target output speed within the specified tolerance. To achieve this, sprocket tooth numbers are first determined based on the desired speed ratio. Subsequently, suitable chain types are evaluated using catalogue power ratings and service factors. Finally, chain length, number of links, lubrication method, and packaging requirements are determined to complete the design.

Throughout the report, catalogue data are used as verification tools to support design decisions rather than as direct selection shortcuts. This approach ensures that the final chain drive configuration satisfies both performance requirements and practical design constraints.

2. Given Data and Design Specifications

The driving unit delivers a mechanical power of 11 kW at a rotational speed of 49 rpm. The driven shaft is required to operate at a nominal speed of 35 rpm, with an allowable deviation of ±7%, which defines the acceptable output speed range. This constraint plays a key role in determining the sprocket tooth numbers and the transmission ratio.

The system is intended to operate for 20 hours per day, indicating a relatively long service duration. Therefore, service conditions and corresponding correction factors must be taken into account during chain selection to ensure sufficient durability and safety.

An approximate center distance of 1616 mm between the driving and driven shafts is specified. This geometric limitation affects the chain length and the total number of links and is checked against recommended catalogue limits during the design process.

All calculated input values are summarized below and used as the basis for subsequent design steps:

- **Transmitted Power (P):** 11 kW
- **Driving Shaft Speed (n_1):** 49 rpm
- **Driven Shaft Speed (n_2):** 35 rpm ± 7%
- **Service Time:** 20 h/day

- **Approximate Centre Distance (C):** 1616 mm

These parameters define the operational boundaries of the chain drive system. In the following sections, they are used to determine the required transmission ratio, sprocket tooth numbers, and suitable chain type in accordance with catalogue recommendations.

3.Calculations

3.1Single Standard Chain

The system operates under moderate shock conditions and is driven by an electric motor. According to these conditions, an application factor of 1.3 is used in the design. As no temperature data is given, standard operating conditions are assumed and the temperature factor is taken as 1. Using the selected correction factors, the design power of the system is calculated.

Table 1

Application service factor (F_a)

Load classification	Driven equipment	Type of prime mover: Electric motor or turbine	Internal combustion engine > 6 cylinders, with flywheel, or hydraulic coupling	Internal combustion engine <6 cylinders, with NO flywheel, or hydraulic coupling
Uniform load (U)	Agitators; centrifugal blowers; generators, centrifugal pumps; Uniformly loaded belt conveyor, lightly loaded chain conveyors	1.0	1.0	1.2
Moderate shock (M)	Centrifugal compressors,; kilns and dryers; conveyors and elevators with intermittent, medium load fluctuations; Dryers; Pulverisers; machinery with moderate pulsating loads (machine tools paper, textiles)	1.3	1.2	1.4
Heavy shock (H)	Press, construction and mining equipment; reciprocating machinery, (compressors, reciprocating feeders, oil well rigs) rubber mixers, roll lines, machinery with heavy shock or reversing torques	1.5	1.4	1.7 – 1.9

Figure 1: Service Factor Table

Table 3

Temperature factor (F_T)

Temperature range °C	F_T factor
–40 °C to –30 °C	Not suitable
–30 °C to –20 °C	0.25
–20 °C to –10 °C	0.33
–10 °C to +150 °C	1.00
150 °C to +200 °C	0.75
+200 °C to +250 °C	0.5
> +250 °C	Refer SKFPTP

Figure 2: Temperature Factor

$$F_a = 1.3$$

$$F_T = 1$$

$$P_d = P_n * F_a * F_T$$

$$P_d = 11 * 1.3 * 1 = 14.3\ kW$$

$$Sprocket\ Ratio\ (i) = \frac{Driver\ Speed\ (n_1)}{Driven\ Speed(n_2)}$$

$$i = \frac{49\ rpm}{35\ rpm} = 1.4$$

Table 4

BS / DIN preferred sprockets

11	*12*	*13*	15	17	19	20	21	23	25	27	30
38	45	57	76	95	114						

ANSI preferred sprockets

9	*10*	*11*	*12*	*13*	14	15	16	17	18	19	20
28	30	32	35	36	40	42	45	48	52	54	60
70	72	80	84	96	112						

Figure 3: Sprocket Teeth Number Table

Sprocket tooth numbers were selected from the BS/DIN preferred sprocket table to ensure smooth chain operation and compatibility with standard components. For the driving sprocket, several preferred values such as 17, 19, 21, and 23 teeth were considered.

Smaller tooth numbers were not selected because they increase polygonal action and chain articulation, which may lead to higher dynamic loads and wear. Larger values (above 21 teeth) were also avoided, as they would require a correspondingly larger driven sprocket to achieve the same transmission ratio, resulting in increased overall size without a clear operational benefit.

Based on these considerations, 19 teeth was selected for the driving sprocket as a balanced choice.

Using the target transmission ratio of approximately 1.40, the driven sprocket tooth number was then determined from the same preferred list. The closest suitable value was found to be 27 teeth, resulting in a transmission ratio of:

$$i = \frac{27}{19} = 1.421$$

This ratio yields a driven speed that remains within the allowable ±7% tolerance range. Other combinations either produced ratios outside the acceptable speed range or led to unnecessarily larger sprocket sizes. Therefore, the sprocket pair 19/27 was selected for the final design.

32B-1; (50.80 mm Pitch) Power ratings in kilowatt (European standard)

No of teeth	Pitch circle Dia.	rpm of small (faster) sprocket z_1															
Z	mm	10	25	50	75	100	125	150	175	200	225	250	300	350	400	450	500
13	**212,27**	3,47	8,03	14,85	21,30	27,55	32,46	40,09	45,13	51,32	56,79	62,83	73,98	85,15	96,13	103,96	116,73
15	**244,33**	4,04	9,35	17,25	22,59	32,19	34,42	46,27	47,86	59,92	60,24	73,39	86,70	99,57	112,45	110,26	137,33
17	**276,46**	4,64	10,56	19,75	27,75	36,82	42,29	52,96	58,81	68,67	74,01	83,95	99,57	113,30	128,75	135,47	157,08
19	**308,64**	5,22	12,02	$\boxed{22,32}$	32,27	41,54	49,17	59,66	68,38	77,43	86,06	94,42	111,58	128,75	145,05	157,52	177,68
21	**340,84**	5,81	13,31	24,80	35,49	46,35	54,09	66,52	75,22	86,70	94,67	105,58	124,45	142,48	161,37	173,27	197,42
23	**373,07**	6,42	14,76	27,38	38,73	51,07	59,01	73,39	82,06	95,28	103,27	115,88	137,33	157,93	177,68	189,03	218,02
25	**405,32**	7,02	16,14	30,05	41,95	55,88	63,92	80,34	88,90	103,85	111,88	127,90	150,20	172,53	193,98	204,84	235,18
Lubrication method		**TYPE 1**		**TYPE 2**						**TYPE 3**							

Figure 4: Power rating table of 32B-1

The selection of the chain size was performed using the BS/DIN single-strand (simplex) power rating tables provided in the SKF catalogue. Since the initial design stage considers a single-strand chain, only tables corresponding to B-1 chain types were evaluated.

The power rating tables are organized as a function of the small sprocket tooth number and the rotational speed of the driving shaft. Therefore, the previously selected small sprocket tooth number ($Z_1 = 19$) and the operating speed of approximately 50 rpm were used as fixed reference parameters when reading the tables.

For each candidate B-1 chain size, the rated power value was obtained from the intersection of the $Z_1 = 19$ row and the 50 rpm column. This rated power was then compared with the required design power $P_d = 14.3$ kW.

According to the design criterion, the selected chain must satisfy:

$$P_{rated} \geq P_d$$

Following the smallest suitable chain principle, the power rating tables were checked starting from smaller chain sizes and progressing to larger ones. The first chain size for which the rated power exceeded the design power was selected as the final single-strand chain. Larger chain sizes were not selected, as they would increase weight and size without providing a necessary performance advantage.

Based on the selected sprocket tooth numbers and chain parameters, the pitch diameters of both sprockets can be determined. For the small sprocket, the pitch diameter is obtained directly from the corresponding catalogue figure. According to this reference, the pitch diameter of the small sprocket is 308.64 mm, and the associated chain pitch is 50.80 mm.

The pitch diameter of the large sprocket is calculated analytically using the standard geometric relationship between chain pitch and sprocket tooth number. The following expression is used to determine the pitch diameter of the large sprocket:

$$D = \frac{\text{pitch}}{\sin\left(\frac{180}{T_2}\right)}$$

where pitch represents the chain pitch length and T_2denotes the number of teeth on the large sprocket. Using this equation, the pitch diameter of the large sprocket is calculated based on the selected chain pitch and tooth number.

$$D = \frac{50.80}{\sin\left(\frac{180}{27}\right)} \approx 437.6 \text{ mm}$$

Another parameter that influences the design power of the chain drive is the speed factor, which depends on the linear speed of the chain. Therefore, the chain speed must be calculated before selecting the corresponding speed factor from the catalogue.

The chain speed is determined using the pitch diameter of the small sprocket, since it is connected to the driving shaft. The chain speed is calculated as:

$$v = \frac{\pi\, d\, n_1}{60}$$

where
d is the pitch diameter of the small sprocket (mm),
n_1 is the rotational speed of the driving shaft (rpm).

Substituting the calculated values:

$$v = \frac{\pi \times 308.64 \ \times 49}{60}$$

$$v = 791.5 \text{ mm/s} = 0.792 m/s$$

Since the chain forms a closed loop, its linear speed is identical at both the driving and driven sprockets. This can be verified using the driven sprocket parameters:

$$v = \frac{\pi\, D\, n_2}{60}$$
$$v = \frac{\pi \times 437.6 \times 34.5}{60}$$
$$v = 791.5 \text{ mm/s} = 0.792 m/s$$

Thus, the chain speed calculated at the small and large sprockets is consistent, confirming the validity of the selected sprocket dimensions.

			Table 2
Speed of chain	Speed factor	Speed of chain	Speed factor
m/s	F_n	m/s	F_n
Less than 0.17	1.0	>0.5 – < 0.67	1.3
>0.17 and <0.33	1.1	>0.67 – < 0.83	1.4
>0.33 and < 0.5	1.2	>0.83 – <1.17	1.6

Figure 5: Table of Speed Factor

The design power represents the effective power that the chain drive must be capable of transmitting under actual operating conditions. It is obtained by applying correction factors to the nominal motor power in order to account for service conditions, chain speed, and operating temperature.

$$P_d = P \cdot F_a \cdot F_n \cdot F_t$$

$$Pd = 11 \times 1.3 \times 1.4 \times 1.0$$

$$Pd = 20.02\ kW$$

The design power accounts for the actual operating conditions of the system by including the effects of load characteristics and chain speed. This value provides a safe and realistic basis for selecting a suitable chain size for the given application.

The chain length is determined based on the selected sprocket tooth numbers and the center distance expressed in pitch units. A correction factor is applied to account for the difference between the sprocket sizes before calculating the final number of links.

L: Chain length (in pitches)

Z1: Number of teeth of small sprocket

Z2: Number of teeth of large sprocket

C: Shaft centers (in pitches)

K: A constant, determined from (Z2–Z1), in Table 8 at catalogue.

Table 8

'K' Factors

Z_1-Z_2	K	Z_1-Z_2	K	Z_1-Z_2	K	Z_1-Z_2	K	Z_1-Z_2	K	Z_1-Z_2	K	Z_1-Z_2	K
1	0,0	11,00	3,06	21,00	11,17	31,00	24,34	41,00	42,58	51,00	65,88	61	94,25
2	0,1	12,00	3,65	22,00	12,26	32,00	25,94	42,00	44,68	52,00	68,49	62	97,37
3	0,2	13,00	4,28	23,00	13,40	33,00	27,58	43,00	46,84	53,00	71,15	63	100,54
4	0,4	14,00	4,96	24,00	14,59	34,00	29,28	44,00	49,04	54,00	73,86	64	103,75
5	0,6	15,00	5,70	25,00	15,83	35,00	31,03	45,00	51,29	55,00	76,62	65	107,02
6	0,9	16,00	6,48	26,00	17,12	36,00	32,83	46,00	53,60	56,00	79,44	66	110,34
7	1,2	17,00	7,32	27,00	18,47	37,00	34,68	47,00	55,95	57,00	82,30	67	113,71
8	$\boxed{1,6}$	18,00	8,21	28,00	19,86	38,00	36,58	48,00	58,36	58,00	85,21	68	117,13
9	2,1	19,00	9,14	29,00	21,30	39,00	38,53	49,00	60,82	59,00	88,17	69	120,6
10	2,5	20,00	10,13	30,00	22,80	40,00	40,53	50,00	63,33	60,00	91,19	70	124,12
71	127,7	81,00	166,19	91,00	209,76	101,00	258,39	111,00	312,09	121,00	370,86	131	434,69
72	131,3	82,00	170,32	92,00	214,40	102,00	263,54	112,00	317,74	122,00	377,02	132	441,36
73	135,0	83,00	174,50	93,00	219,08	103,00	268,73	113,00	323,44	123,00	383,22	133	448,07
74	138,7	84,00	178,73	94,00	223,82	104,00	273,97	114,00	329,19	124,00	389,48	134	454,83
75	142,5	85,00	183,01	95,00	228,61	105,00	279,27	115,00	334,99	125,00	395,79	135	461,64
76	146,3	86,00	187,34	96,00	233,44	106,00	284,67	116,00	340,84	126,00	402,14	136	468,51
77	150,2	87,00	191,73	97,00	238,33	107,00	290,01	117,00	346,75	127,00	408,55	137	475,42
78	154,1	88,00	196,16	98,00	243,27	108,00	295,45	118,00	352,70	128,00	415,01	138	482,39
79	158,1	89,00	200,64	99,00	248,26	109,00	300,95	119,00	358,70	129,00	421,52	139	489,41
80	162,1	90,00	205,18	100,00	253,30	110,00	306,50	120,00	364,76	130,00	428,08	140	496,47

Figure 6: K factor for Chain Length

$$L = \frac{Z_2 + Z_1}{2} + 2C + \frac{K}{C}$$

$$C = \frac{C_{mm}}{p} = \frac{1616}{50.80} = 31.81$$

$$L = \frac{27 + 19}{2} + 2(31.81) + \frac{1.6}{31.81}$$

$$L = 23 + 63.62 + 0.05 = 86.67$$

$$L = 88\ links$$

After calculating the chain length, the result was obtained as a non-integer value. Since roller chains must be assembled with an even number of links, the chain length was rounded to the nearest feasible even values. Among the possible options, the selected chain length provides a reasonable balance between chain tension and sag. A shorter chain would result in higher initial tension and more difficult assembly, while a longer chain would introduce excessive slack and reduce positional accuracy. Therefore, the selected number of links was considered the most suitable choice for this design.

$$L_{mm} = 88 \times 50.8 = 4470.4 \text{ mm}$$

$$L = 4.47m$$

According to the catalogue, chains are supplied in standard boxes of 5 m length. Therefore, the number of required boxes is calculated as:

$$N = \frac{L_{required}}{L_{box}} = \frac{4.47}{5} = 0.89$$

Since partial boxes cannot be purchased, the required number of boxes is rounded up to the nearest integer:

$$\boxed{N = 1 \text{ box}}$$

Based on the selected chain type (32B-1) and the corresponding power rating table, Type 1 lubrication was selected. Since the calculated chain speed lies within the range covered by the Type 1 lubrication recommendation in the catalogue, this method is sufficient to ensure proper operation and acceptable wear under the given working conditions.

3.2Double Strand Chain

The double-strand chain configuration is evaluated using the same operating conditions and design parameters defined for the single-strand chain. The application factor, speed ratio, and sprocket tooth numbers are therefore unchanged. This approach allows the effect of using multiple strands to be assessed without altering the geometric or operating constraints of the system.

Application factor: $F_a = 1.3$
Sprocket tooth numbers: 19T–27T
Design power (without speed factor): $P_d = 14.3$ kW

28B-1; (44.45 mm Pitch) Power ratings in kilowatt (European standard)

No of teeth	Pitch circle Dia.	rpm of small (faster) sprocket z_1															
Z	mm	10	25	50	75	100	125	150	200	250	300	350	400	450	500	550	600
13	**185,74**	2,63	5,99	11,25	14,97	20,94	22,82	30,05	38,96	47,64	55,96	64,38	72,53	80,68	89,27	97,00	87,55
15	**213,79**	3,06	7,00	13,05	15,87	24,46	24,21	35,10	45,50	55,62	65,49	75,37	84,98	94,42	103,85	113,30	108,15
17	**241,91**	3,52	8,00	14,94	19,50	27,89	29,74	40,17	52,02	63,52	75,11	86,70	97,00	108,15	129,60	129,60	129,60
19	**270,06**	3,96	9,01	$\boxed{16,48}$	22,67	31,42	34,58	45,32	58,79	72,10	84,20	97,00	109,00	121,88	145,92	145,92	145,92
21	**298,24**	4,41	10,12	18,72	24,94	35,02	38,04	50,47	65,23	79,83	94,42	108,15	121,88	135,62	163,95	163,95	175,95
23	**326,44**	4,87	11,15	20,68	27,20	38,63	41,49	55,80	72,10	88,40	103,85	118,45	134,75	149,35	179,40	179,40	193,98
25	**354,65**	5,33	12,18	22,66	29,47	42,41	44,95	60,95	78,97	97,00	113,30	130,47	147,63	163,95	204,28	204,28	212,00
Lubrication method		TYPE 1		TYPE 2								TYPE 3					

Figure 7: Power Rating Table for 28B-1

For the double-strand chain design, type 28B was examined. Using the power rating table for Z₁ = 19 and an operating speed of approximately 50 rpm, the rated power of the 28B single-strand chain was obtained as 16.48 kW. This value is used as the basis for evaluating the suitability of the chain before considering the effect of multiple strands.

Table 6

Multiple strand factor

No. strands	Multiplier K2	No. strands 1	Multiplier K2
1	1.0	4	3.3.
2	$\boxed{1.7}$	5	3.9
3	2.5	6	4.6

Figure 8: Multiple Strand Factor

$$P_{rated,2} = 1.7 \times 16.48 = 28.016 \text{ kW}$$
$$\boxed{P_{rated,2} \approx 28.02 \text{ kW}}$$

After selecting the sprocket tooth numbers and the chain pitch, the pitch diameter of the large sprocket can be determined. The pitch diameter is a geometric quantity that relates the chain pitch to the sprocket tooth count and is required for checking chain speed and overall drive geometry. For a sprocket with T_2teeth and chain pitch p, the pitch diameter is calculated by:

$$D = \frac{p}{\sin\left(\frac{180}{T_2}\right)}$$

Now, using your values for the double-strand 28B chain:

- $p = 44.45$ mm
- $T_2 = 27$

$$D = \frac{44.45}{\sin\left(\frac{180}{27}\right)}$$

$$D = \frac{44.45}{0.1161} = 382.88 \text{ mm}$$
$$D = 382.88mm$$

The chain speed is the linear velocity of the chain along the sprockets. It is calculated to select the speed factor from the catalogue and to verify that the operating conditions fall within the recommended lubrication range. Since the chain is driven by the small sprocket, the chain speed is computed using the small sprocket pitch diameter and the driving shaft speed.

$$v = \frac{\pi d n_1}{60}$$

Using your double-strand (28B) values:

- $d = 270.06$ mm
- $n_1 = 49$ rpm

$$v = \frac{\pi \times 270.06 \times 49}{60} = 693.9 \text{ mm/s} = 0.694 \text{ m/s}$$

So, the chain speed is:

$$v = 0.694 \text{ m/s}$$

Since the chain forms a continuous loop, its linear speed is the same at both the small and large sprockets. This can also be verified using the pitch diameter of the large sprocket and the driven shaft speed:

$$v = \frac{\pi\, D\, n_2}{60}$$

Using:

- $D = 382.88$ mm
- $n_2 = 34.5$ rpm

$$v = \frac{\pi \times 382.88 \times 34.5}{60}$$
$$v = 693.9 \text{ mm/s} = 0.694 \text{ m/s}$$

Thus, the chain speed calculated at both sprockets is consistent, and this value is used to determine the corresponding speed factor from the catalogue.

According to the sprocket ratio, the actual driven speed is calculated as:

$$n_2 = \frac{n_1 Z_1}{Z_2} = \frac{49 \times 19}{27} = 34.48 \text{ rpm}$$

Based on the calculated chain speed ($v = 0.694$ m/s) and the speed factor table, the speed factor is selected as:

$$F_n = 1.4$$

Therefore, the design power is recalculated by including all correction factors:

$$P_d = P \cdot F_a \cdot F_t \cdot F_n$$
$$P_d = 11 \times 1.3 \times 1.0 \times 1.4$$
$$P_d = 20.02 \text{ kW}$$

The recalculated design power is checked against the catalogue power rating to confirm chain suitability. Since the rated capacity is higher than the design power, the selected chain is suitable. After finalizing the design power, the next step is to determine the chain length.

$$Z_1 = 19, Z_2 = 27$$
$$p = 44.45 \text{ mm}$$
$$C = \frac{C_{mm}}{p} = \frac{1616}{44.45} = 36.36$$
$$K = 1.6$$
$$L = \frac{27 + 19}{2} + 2(36.36) + \frac{1.6}{36.36}$$
$$L = 23 + 72.72 + 0.04$$
$$L = 95.76 \text{ pitches}$$

Since the number of links must be even, the chain length is selected as:

$$L = 96 \text{ pitches}$$
$$L_{mm} = p \cdot L = 44.45 \times 96 = 4267.2 \text{ mm}$$
$$L = 4.27 \text{ m}$$

To calculate the required number of boxes, the standard supply length of the chain must be considered. The calculated chain length for this design is $L \approx 4.27$ m. Since the chain is supplied in 5 m boxes, one box is sufficient:

$$N = \frac{L_{required}}{L_{box}} = \frac{4.27}{5} = 0.85 = 1\ box$$

Therefore, the selected chain designation for the double-strand option is:

28B-2 (double-strand roller chain), 5 m box

Finally, the lubrication method is selected from the lubrication recommendation shown under the corresponding 28B power rating table. For the operating conditions of this system, the recommended lubrication type is Type 2.

4.Conclusion

In this study, a chain drive system for a centrifugal compressor was designed in accordance with the specifications given by instructor and by using the SKF Power Transmission Chain Catalogue. The design process was primarily governed by the required output speed, which was defined as the most critical design criterion. Based on the given input data, a suitable speed ratio was achieved by selecting standard DIN sprocket tooth numbers of 19 and 27, resulting in an output speed within the allowable tolerance range. The approximate center distance provided in the problem statement was preserved, and all geometric and kinematic constraints were satisfied.

Two alternative chain drive configurations were evaluated in order to compare different design approaches. For the single-strand configuration, a DIN 32B-1 roller chain was selected after calculating the design power by applying the application, speed, and temperature correction factors. The selected chain was verified using the power rating tables, and the corresponding sprocket dimensions, chain speed, chain length, lubrication method, and packaging requirements were determined.

In addition to the single-strand design, a double-strand (duplex) chain configuration was examined as an alternative solution. By applying the multiple strand factor, it was shown that a DIN 28B-2 roller chain is capable of transmitting the required design power while maintaining the same sprocket geometry and center distance. This option provides increased power capacity through load sharing between the strands and offers a compact alternative compared to the single-strand design.

For both configurations, the required chain length was calculated and adjusted to an even number of pitches to ensure proper assembly. Since the calculated chain lengths are shorter than the standard supply length, one 5 m chain box is sufficient for each design. The lubrication methods were selected based on the recommendations provided in the corresponding power rating tables, ensuring reliable operation under the given working conditions.

The required parts to be ordered for the selected chain drive systems are:

- One driving sprocket with 19 teeth,
- One driven sprocket with 27 teeth,

- One roller chain (DIN 32B-1 for the single-strand option or DIN 28B-2 for the double-strand option),
- One standard 5 m chain box.

During installation, accurate alignment of the sprockets and correct center distance adjustment are essential to prevent excessive wear and vibration. The chain should be installed with the calculated number of links and adjusted to achieve proper tension without excessive slack. During operation, the recommended lubrication method must be applied according to the catalogue guidelines, and periodic inspections should be carried out to ensure safe and reliable long-term operation.

Overall, the final designs satisfy all specified design requirements and demonstrate feasible and reliable solutions for the given application, providing flexibility in chain selection depending on performance and design preferences.